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THE RIGID AIRSHIP *ZR-3*.

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HISTORY OF HER ACQUISITION BY THE UNITED STATES.

Under the terms of the Treaty of Peace with Germany, two of the existing German rigid airships were to be turned over to the United States. These ships were destroyed by their own personnel, somewhat under the same circumstances that prompted the sinking of the surrendered German warships at Scapa Flow. The German Government was required to replace the destroyed airships by a payment either in money or kind. It was agreed between the American and German Governments that the payment should be made by supplying a new airship, estimated to be equivalent in value to the two old ones prior to

their destruction. The Council of Ambassadors ruled that this airship should be constructed and used strictly for commercial purposes, and that the gas capacity should not exceed 70,000 cubic meters (2,470,000 cubic feet), or approximately the same as the largest airship previously constructed in Germany. It was further provided that after the delivery of this airship to the United States, the shed at Friedrichshafen, in which she has been constructed, must be torn down in accordance with the Treaty of Versailles; and thereafter no airship exceeding 30,000 cubic meters volume may be built in Germany.

In spite of the insistence upon the strictly commercial uses of the new airship, it was arranged that the design and construction should be carried out under the inspection and to the approval of a United States Naval Inspector with offices at the plant of the Zeppelin Company at Friedrichshafen. The agreement between the American and German Governments provided also that delivery of the ship was to be made at the risk of the German Government to the United States Naval Air Station, Lakehurst, N. J.

It may be stated as a matter of interest that the German Government is paying the Zeppelin Company the sum of approximately 3,000,000 gold marks ($750,000) for the construction and delivery of the *ZR-3,* but the United States is in no way concerned with or responsible for this payment. A supplementary contract amounting to $150,000 has been entered into between the United States Government and the Zeppelin Company for supplying certain accessories and spare parts with the ship, and for a course of instruction and training to American officers at Lakehurst after her delivery.

The official American designation, *ZR-3,* is in line with the letters and numerals previously adopted for U. S. Naval rigid airships. The *ZR-1,* commissioned U. S. S. *Shenandoah,* is the first, and at present the only, rigid airship constructed in America. The *ZR-2* was the ill-fated airship constructed in England, commenced for the Air Ministry as the *R-38,* sold to the United States Navy during construction, and destroyed by

breaking in two and subsequent fire and explosion while making her fourth trial trip in England, with the loss of 44 American and British officers and men.

The builder's number of the *ZR-3* is *LZ-126*, signifying the 126th Zeppelin airship.

COMPARATIVE CHARACTERISTICS OF THE "ZR-1" AND "ZR-3."

The dimensions, weights, and performance characteristics of the *ZR-1* (*Shenandoah*) and the *ZR-3* are shown in the following tables:

GENERAL CHARACTERISTICS OF AIRSHIPS "ZR-1" AND "ZR-3."

	ZR-1	*ZR-3*
Air displacement of hull, cubic feet	2,289,861	2,764,461
Volume of gas cells, cubic feet	2,115,174	2,599,110
Length, feet and inches	680–2	658–4
Maximum diameter, feet and inches	78–9	90–8
Total height, feet and inches	93–2	104–5
Engines	5 Packard	5 Maybach
Total H. P.	1500	2000
Areas of tail surfaces:		
Horizontal (2), square feet	2870	2510
Vertical (2), square feet	2335	2510
Areas of rudders:		
Horizontal (2), square feet	446.2	457
Vertical (2), square feet	379.1	400
Area of lateral plane, square feet	42,300	46,800
Area of cross section, square feet	4,818	6,422

COMPARISON OF PRINCIPAL WEIGHTS,

95 per cent Inflated with Helium Lifting .064 pound per cubic foot.

Items	*ZR-1*		*ZR-3*	
	Weight Pounds	Per cent Gross Lift	Weight Pounds	Per cent Gross Lift
Fixed weights	80,226	62.2	91,030	57.6
Non-disposable weights	20,539	15.9	25,200	16.0
Disposable weights	28,235	21.9	41,770	26.4
Total weights	129,000	100.0	158,000	100.0

COMPARISON OF FIXED WEIGHTS.

Items	*ZR-1* As built 1/3/23		*ZR-3* Estimated	
	Weight	Per cent Gross Lift	Weight	Per cent Gross Lift
Hull Structure (not otherwise included)	31,715	23.20	34,170	20.25
Outer Cover, Gas Cells, Valves and Trunks	16,891	12.37	17,200	10.22
Power Plant, wet. (including cars)	20,204	14.80	19,840	11.78
Fuel System (*ZR-1* 62 tanks *ZR-3* 70 tanks)	2785	2.04	4400	2.61
Ballast Bags (for maximum capacity)	696	0.51	660	0.39
Control System—including rudders	1965	1.44	1540	0.92
Navigating Instruments, Electric and Radio Equipment	1922	1.41	2640	1.57
*Landing Equipment (Bow mooring and Handling)	1791	1.31	1320	0.78
Compartments in Corridor (Quarters)	670	0.49	3090	1.83
Control and Passenger Car, including Ventilating Equipment	1587	1.16	6170	3.66
Total fixed weight	80,226	58.73	91,030	54.01
Gross Lift (100 per cent full of hydrogen)	136,634	.....	168,300	.....

NOTE:

ZR-1 Fuel tank capacity provided for 37,500 pounds of fuel or 27.5 per cent of gross lift.

ZR-3 Fuel tank capacity provided for 46,200 pounds of fuel, or 27.5 per cent of gross lift.

Normal ballast capacity on *ZR*-1 provided for 14,300 pounds of ballast, or 10.5 per cent of gross lift.

Maximum ballast capacity on *ZR*-1 provided for 28,600 pounds of ballast, or 21.0 per cent of gross lift.

Normal ballast capacity on *ZR*-3 provided for 17,500 pounds of ballast, or 10.5 per cent of gross lift.

Maximum ballast capacity on *ZR*-3 provided for 35,200 pounds of ballast, or 21.0 per cent of gross lift.

*Allowance for Bow Mooring equipment and structure not included in fixed weight of *ZR-3*.

The non-disposable weights include the crew, emergency rations, fuel and oil for 5 hours at cruising speed, and minimum ballast. The disposable weights represent the maximum fuel and oil, over and above the 5 hours reserve, which may be carried in the given conditions.

It may be noted that the *ZR-3* is shorter and of greater diameter than the *ZR-1*. Contrary to what might at first be supposed, the shorter and fatter form of the new ship, coupled with the elimination of parallel middle body, offers relatively less resistance than the more slender hull of the *Shenandoah*, due chiefly to reduction in surface area and frictional resistance. In fact, experiments on airship models in wind tunnels have shown that the least resistance for a hull of a given volume is obtained with a length/diameter ratio below five, although for various reasons it appears inexpedient at present to construct rigid airships of such dumpy proportions.

SPEED AND ENDURANCE.

The estimated full speed of the *ZR-3* is 65.7 knots, with a fuel consumption of 970 pounds per hour. At two-thirds full power, the estimated speed is 58 knots, and the fuel consumption is 580 pounds per hour. When cruising with 3 engines two-thirds full out, the estimated speed is 48.5 knots, and the fuel consumption is 350 pounds per hour.

When inflated with helium, 95 per cent full, the maximum fuel load which can be carried over and above 5 hours reserve is estimated at 41,770 pounds, whence the endurance is as follows:

	Endurance	
Speed, knots.	*Hours.*	*Nautical miles.*
65.7	43	2825
58.0	72	4180
48.5	119	5770

Fuel consumption trials of the *Shenandoah* have shown the consumption to be 566 pounds per hour at 51 knots, and 292 pounds per hour at 41 knots. When inflated 95 per cent full of

helium, lifting .064 pound per cubic foot, the available weight for fuel over and above the reserve is 28,235 pounds, whence the endurance is:

	Endurance	
Speed, knots.	*Hours.*	*Nautical miles.*
51	50	2550
41	97	3980

This comparison shows the *Shenandoah* in a very unfavorable light compared with the *ZR*-3; but the figures for the *Shenandoah* are from actual service trials, whereas the computations for the *ZR*-3 are based on the builder's estimate of performance in the most favorable conditions, and have yet to be verified in service.

DETAILED DESCRIPTION OF THE HULL.

Although in general features and type of construction the *ZR-3* closely resembles the *Shenandoah*, there are many differences having considerable significance to the expert. The most outstanding difference is that while in both ships the transverse frames are spaced at 5 meters apart, the alternate frames of the *Shenandoah* are main frames with king-post bracing members and cross-wiring to afford transverse rigidity to the hull and to form bulkheads between adjacent gas cells, but only every third frame of the *ZR-3* is a main frame, thus permitting the lengths of the gas cells to be increased from 10 to 15 meters. This change was embodied in the later war-time Zeppelins, and was found to effect an important saving of weight, particularly in the gas cells, without loss of strength, although there is inevitably some loss of safety due to the decreased sub-division of the gas space.

The keel of the *ZR-3* is quite different from earlier Zeppelins and the *Shenandoah*. Hitherto, rigid airships have been constructed with an odd number of sides, having an apex at the top center line of the hull, and a flat bottom forming the base of an internal isosceles triangle-shaped keel or corridor, as in the *Shenandoah* (Figure 1). The cross-section of the

ZR-3 is basically a regular 24-sided polygon, having apexes on both the top and bottom center lines. At the bottom, a departure is made from the regular polygonal section by dropping the bottom apex below the circumscribing circle until the adjacent sides become collinear with the sides next above, so that as seen from the outside, the hull has but 22 sides, with the two lowest ones twice the width of the others, although structurally the 24 sides are retained, since there is a longitudinal girder along the center of each of the two wide sides. A five-sided keel structure is worked into this peculiar arrangement of the bottom of the ship, as shown in Figures 2 and 3. This construction has a number of important advantages over the usual triangular keel, although it involves some loss of gas space. One advantage is that the two upper members of the keel serve not only as main strength members, but also carry the local loads, such as the fuel tanks, ballast bags, etc., so that no special box girders are required for this purpose as in the *Shenandoah* (Figure 4). Even more important is that the bottom center line girder is of exceptional depth and strength, being the most important longitudinal girder in the ship, and at the same time it makes a strong and rigid walkway, instead of the special and rather "spongy" girder which has hitherto fulfilled this function in rigid airships.

The elements of the main transverse frames are diamond-shaped (Figure 5), instead of consisting of a large straight member with a king-post and two inclined bracing members as in the *Shenandoah* (Figure 6). These elements are subjected mainly to compressive loads, and with the diamond arrangement all members are effective in resisting these loads, whereas the inclined members of the elements in the *Shenandoah* contribute but little to the strength, except to provide rigidity.

The intermediate transverse frames are necessarily without cross-wiring because of the presence of the gas cells; but in the *ZR-3* the girders of these frames are made much deeper than hitherto, thus greatly increasing their effectiveness in giving rigidity to the hull and support to the longitudinal girders.

The wire netting for taking the gas pressure between the longitudinals is of a new design in which exceptionally large numbers of the wires are secured directly to the joints of the frames, so as to relieve the longitudinals of bending loads from gas pressure as much as is practicable.

A peculiar feature of the *ZR-3* is that there is no axial cable to reinforce the wiring of the main transverse frames against gas pressure loads resulting from differences of pressure in adjacent gas cells. An equalization ring is provided in the center of each main frame to give as nearly as possible equal tensions in the diametral wires.

THE LONGITUDINAL GIRDERS.

With the exception of the special deep girders along the top and bottom center lines, the longitudinals of the *ZR-3* are all alike in overall dimensions and functions. The distinction between main and secondary longitudinals, found in the *Shenandoah* and other rigid airships, no longer remains. The history of the development of the longitudinal girders is interesting. In the first Zeppelin airships, longitudinal strength was provided mainly by a strong triangular keel, apex downward, below the hull. The longitudinal girders within the hull contributed but little to the longitudinal strength, their main function being to take the gas pressure loads and support the outer cover.

Later, the keel was greatly lightened and placed within the interior of the hull, apex upwards; and the longitudinals became of two types, as follows: (*a*) Large main longitudinals which provided the principal part of the longitudinal strength of the hull, as well as taking gas pressure and outer cover loads; (*b*) secondary longitudinals disposed alternately with the main ones, having considerably smaller dimensions, and designed to take only gas pressure and outer cover loads. No shear wires were secured to the secondary longitudinals, so that they could take practically no part in resisting the primary bending of the hull. Still later, shear wires were secured to

the secondary longitudinals, so that there was no longer a distinction in the functions of the main and secondary longitudinals, although the difference in size remained. The *Shenandoah* belongs to this stage of the evolution, although in the lower half of the hull the secondary longitudinals are deeper than in previous practice, and, in fact, have the same depth as the longitudinals of the *ZR-3*. Finally, in the *ZR-3*, the distinction between the two types of longitudinals totally disappears.

TYPES OF GIRDER CONSTRUCTION.

Almost all of the girders of the *ZR-3* are of the well known Zeppelin type, as found in the *Shenandoah*. Generally they are triangular in section and are formed by longitudinal channels at the three corners of the section, each face of the girder being braced by a series of crossed, light, corrugated stampings riveted to the channels and to each other where they cross. There are also some four-sided box girders in which the longitudinal members are of angle instead of channel section.

Some girders of a new type are used in the *ZR-3*, chiefly in the control and passenger car. (See Figure 10.) These girders are of rectangular cross section, and each of the four sides consists of a duralumin sheet having a single row of large diameter lightening holes, and two rows of small holes. The metal around the holes is turned inward so as to give great rigidity to the sheets. Girders of this type may readily be reinforced by longitudinal members of angle section riveted along the corners at the intersections of the plates.

BOW MOORING GEAR.

The *ZR-3* is fitted with a bow mooring cone identical with that of the *Shenandoah*. In other respects the bow mooring gear is also practically the same as in the *Shenandoah*, except in the form of the bow of the ship. The spindle is supported by a reverse curvature of the bow structure outwards, instead of by carrying the longitudinals in a practically straight line to the spindle, as in the *Shenandoah* (Figures 7 and 8).

STERN STRUCTURE.

Aft of about the mid-length of the fins, the cross-sections of the hull are octagonal instead of 22-sided. The fairing in of the hull to effect the change in the number of sides is very neatly done, as may be seen by inspection of Figure 9.

THE FINS AND RUDDERS.

There are four fins and rudders, two placed vertically for directional control, and two horizontally for vertical control, just as in all modern rigid airships. An unusual feature, however, is that all four are identical except for the provision of small balancing areas in the horizontal rudders, but not in the vertical ones. In the *Shenandoah,* the bottom fin and rudder are shallower than the other three, in order to give sufficient clearance from the ground. Clearance for the deep bottom fin of the *ZR-3* is obtained by making the rear car with its pneumatic bumping bag, upon which the ship rests when on the ground, considerably lower than the forward bumping bag, which is secured under the control compartment of the forward car. Thus, when the airship is on the ground, she is down by the head with the stern well up.

The fins are of triangular cross-section and internally braced. As structural units they are quite independent of the hull; and the frame spacing of the hull and fins does not even coincide, the spacing being 3.3 meters in the fins, and 5 meters in the hull. The after ends of the fins are braced by strong cruciform girders through the hull, as in the *Shenandoah;* but a novel feature is the provision of a gas cell aft of the cruciform.

The rudders are of high aspect ratio, in accordance with modern aerodynamic theory, and because of their shallowness and the small moment required to turn them, it has not been deemed necessary to provide balancing areas on the vertical rudders. When the airship is statically out of trim it may be necessary to maintain a nearly continuous force in one direction on the horizontal rudders or elevators; and in order to

relieve the altitude coxswain of strain in such conditions, small triangular balancing areas are provided near the outboard ends of the elevators, corresponding areas being cut out of the horizontal fins.

GAS CELLS.

As in the *Shenandoah*, the gas cells are made of single ply cotton cloth lined with goldbeater's skin to give gas tightness. The construction of these cells differs from those of the *Shenandoah* in that there is no rubber adhesive between the skins and the cloth; and the skins are glued to the cloth by a secret process which has not yet been mastered outside of Germany.

All the gas cells, with the exception of the one at the stern, which is in communication with its immediate neighbor, are fitted with 500 millimeter safety valves, the largest cells being fitted with two valves each. The gas discharged from these valves finds its way through passages between the cells to an upper air space between the cells and the covering, which space is kept ventilated by means of seven cowls formed in the hull. Eight of the cells are, in addition, provided with maneuvering valves which can be operated from the control car.

OUTER COVER.

The hull, the keel passage, and the fins are covered with cotton fabric, which varies in thickness according to the strength required to meet the varying loading at different places. The fabric selvedges are reinforced with bands, into which are inserted the eyes for the lacing cords. These cords are of ramie fiber, prepared by a process designed to take account of the stretching of the fabric covering, the loading of the girders, and the temperature and humidity of the atmosphere existing at the time when the covering is applied and laced up. The fabric covering is proofed several times in order to give it as smooth a surface as possible, and the proofing

solution has mixed with it a proportion of aluminum powder so as to give the finished surface a dull mirror-like appearance as in the *Shenandoah.*

THE PASSENGER AND CONTROL CAR.

The arrangement, construction, and appearance of the combined passenger and control car are well shown in the drawings and illustrations (Figures 10 to 17).

The control compartment is much wider and a little shorter than in the *Shenandoah,* but remains essentially the same in its contents. The rudder control wheel is on the center line in the front of the compartment, and the elevator control wheel is aft on the port side. The controls for the gas and ballast valves are overhead, as usual; and the customary navigational instruments and the engine room telegraphs are disposed around the sides of the compartment. A large chart table is placed on the starboard side.

A soundproof radio room is located on the port side, immediately behind the control compartment.

The passenger accommodation includes five separate compartments, each containing two lower and two upper berths, placed athwartships, as in European sleeping cars. By day, the upper berths are folded down to make backs for the lower berths, which then serve as seats. Each of the five compartments is provided with a large window in three sections, and has a wood flooring covered with felt. A table may be placed between the seats of each compartment.

On the starboard side behind the passengers' compartments, there is a kitchen, lined with aluminum on the walls, ceiling and floor. All cooking is by electricity. The entrance passage to the passenger car is on the starboard side, just forward of the kitchen. When the ship is underway this passage serves as the steward's room.

Ample toilet facilities are provided on the port side near the after end of the cabin.

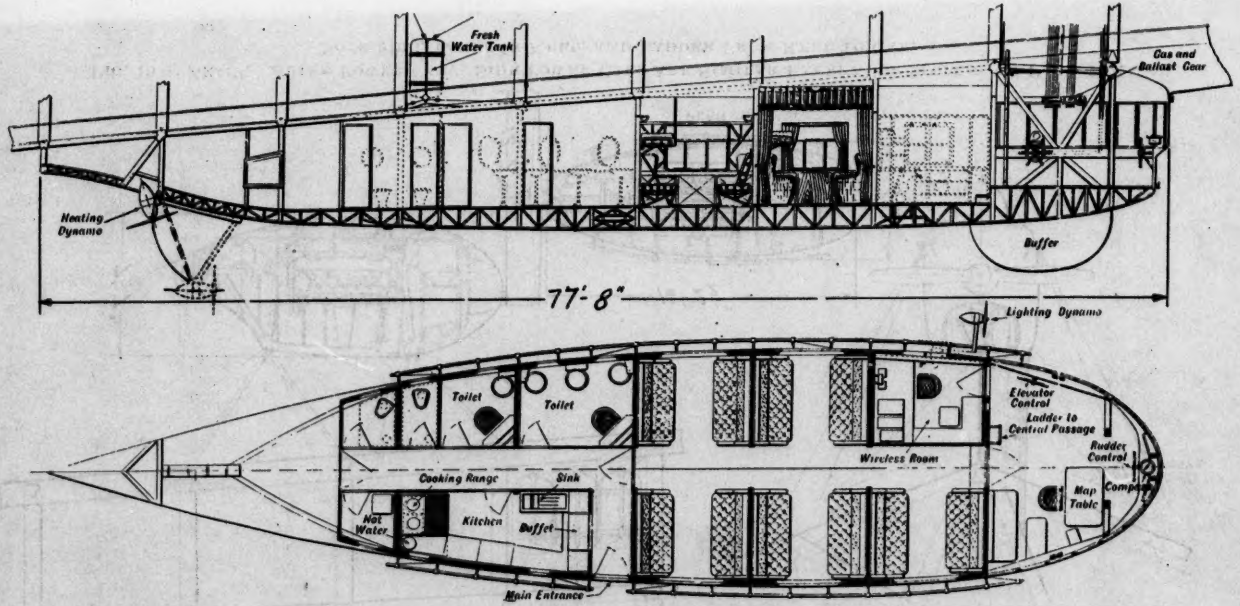


FIG. 17.—"ZR-3". PROFILE AND PLAN OF CONTROL AND PASSENGER CAR. EACH SECTION OF PASSENGER CABIN HOLDS FOUR PERSONS.

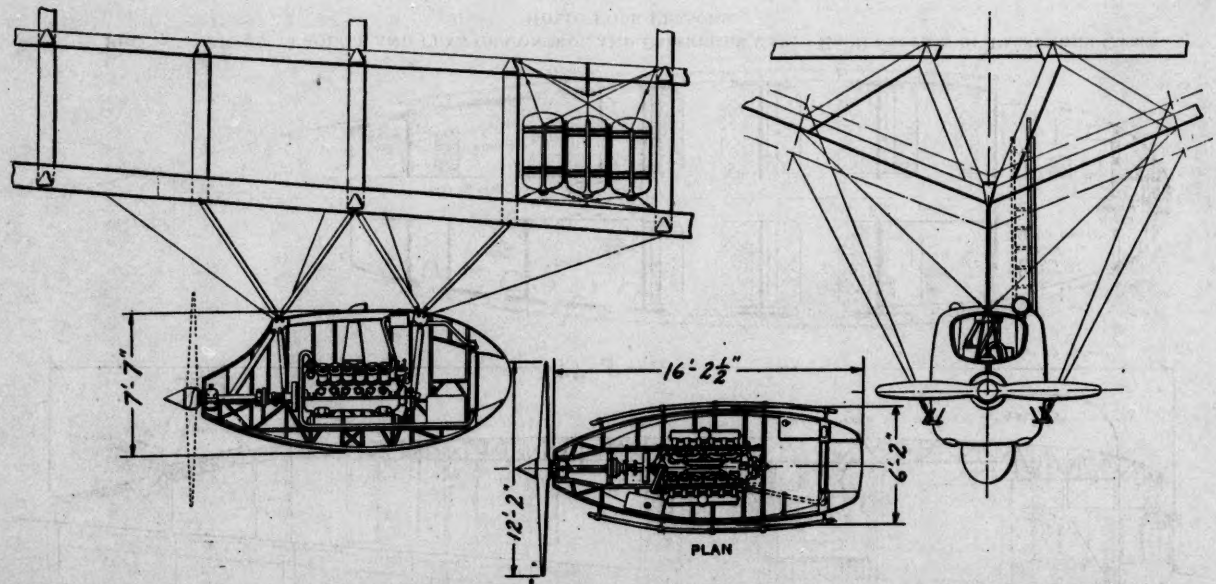
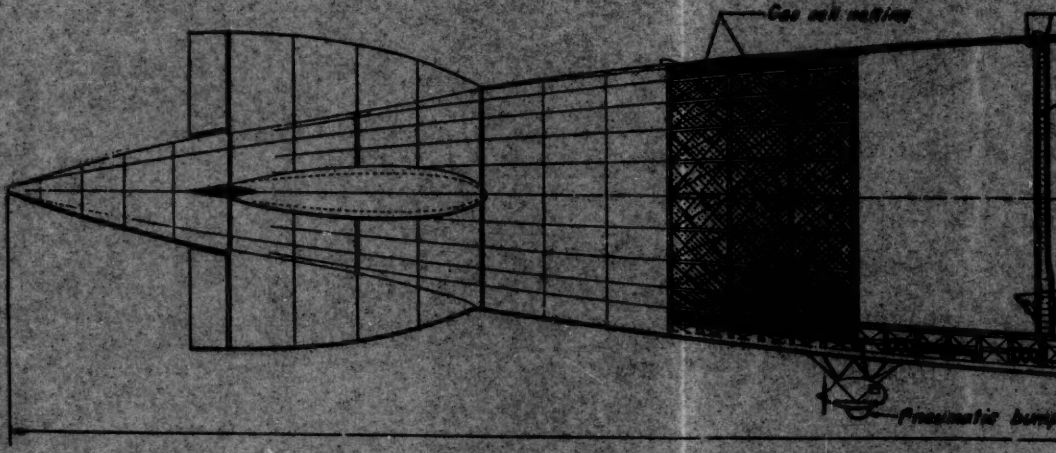
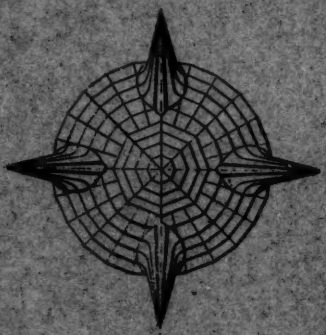
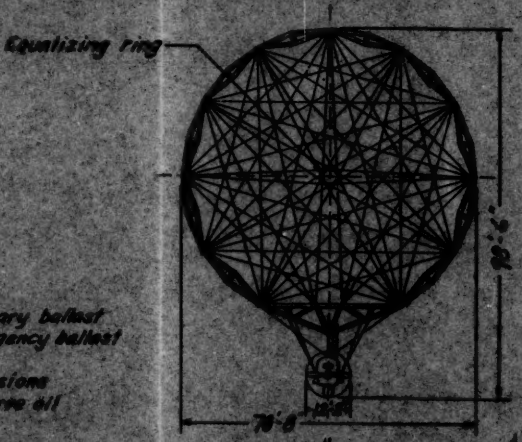


FIG. 18—"ZR-3". AFTER POWER CAR. SIDE POWER CARS ARE SIMILAR EXCEPT FOR SUSPENSION FROM HULL. NOTE FUEL TANKS IN KEEL, AND LADDER FROM KEEL TO CAR.





Stern View



C- Cargo
B- Ordinary ballast
E- Emergency ballast
F- Fuel
P- Provisions
R- Reserve oil

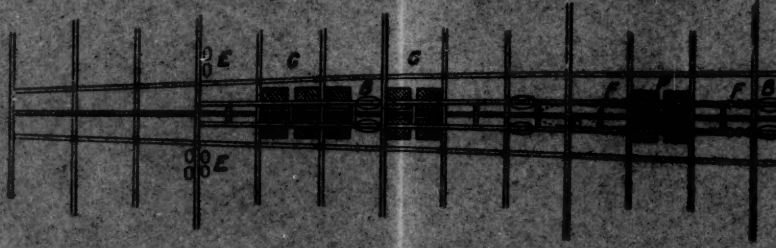


FIG. 19.—"ZR-3." GENER

A." General Arrangement and Cross Sections of Ship, and Layout of Keel.

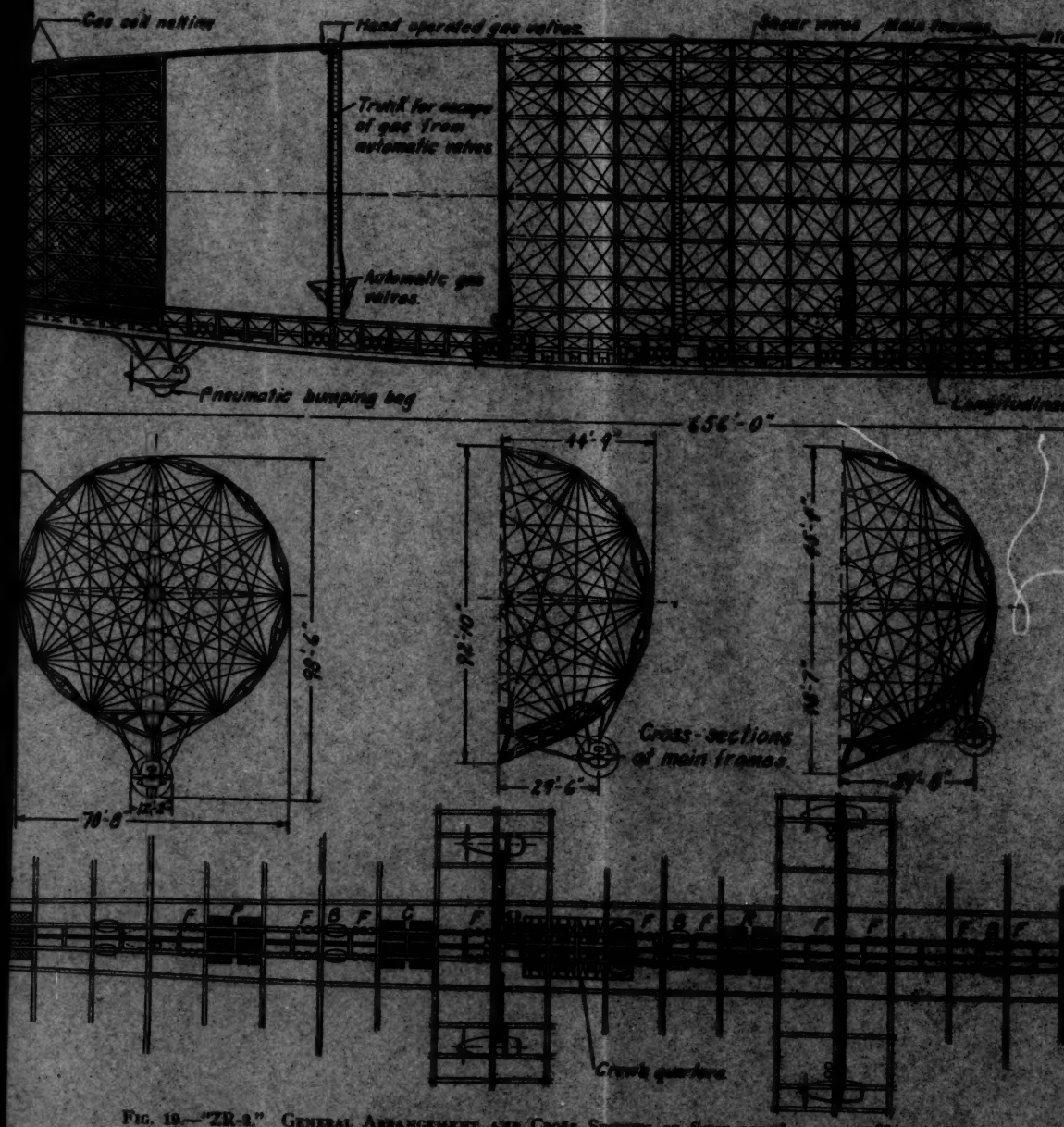
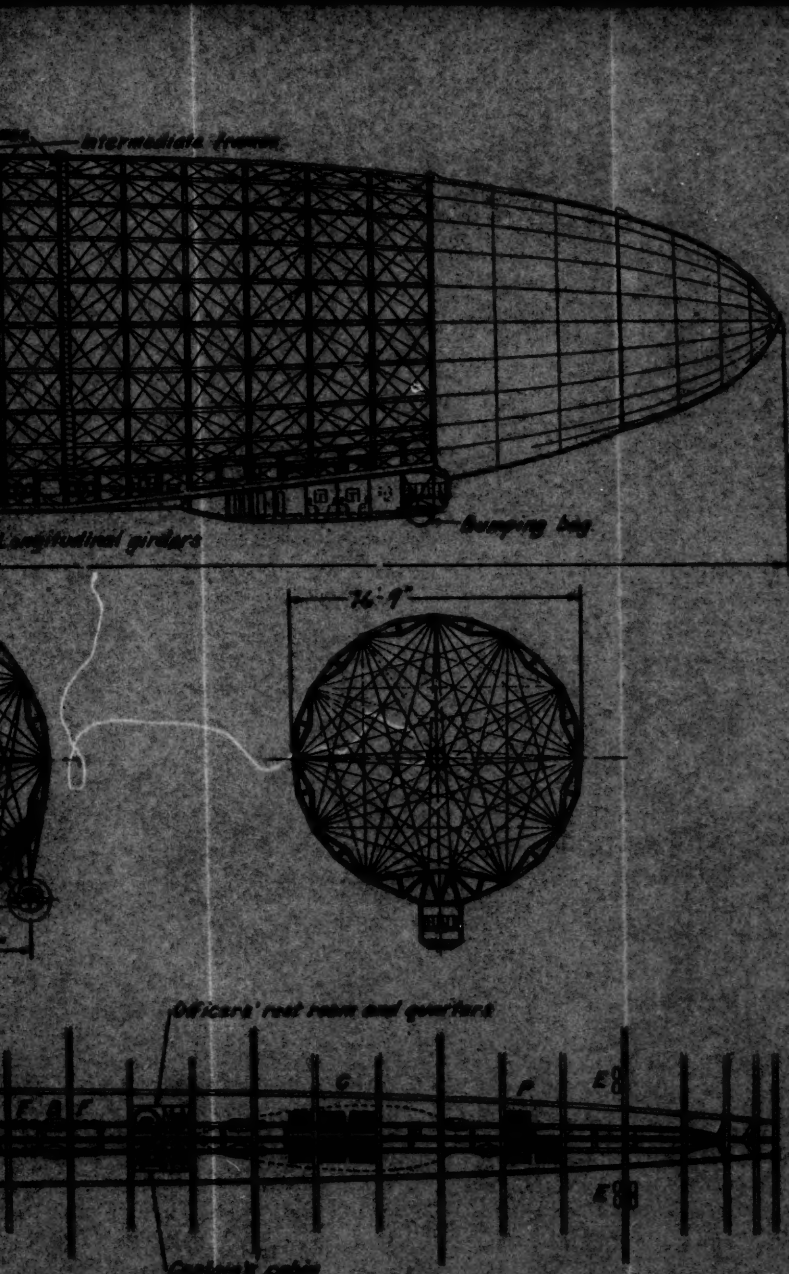


FIG. 19.—"ZR-3." GENERAL ARRANGEMENT AND CROSS SECTION OF SHIP, AND LAYOUT OF KEEL.


intermediate frames
bumping bag
74'-1"
Officers' rest room and quarters
Control cabin

From the technical point of view, the most interesting feature of the combined passenger and control car is the skillful manner in which it has been worked into the hull so as to have very little influence upon the aerodynamic qualities of the ship. In each of the small passenger airships, *Bodensee* and *Nordstern*, constructed by the Zeppelin Company immediately after the Armistice, the passenger and control car was relatively long and narrow, and presented such a large area of vertical longitudinal wall that it acted as a forward fin, seriously interfering with the directional control of the ship, and causing excessive rolling.

In the *Shenandoah* there are no passenger accommodations, and the control car was originally combined with the forward engine car; but the engine was subsequently removed from this car and replaced by an exceptionally powerful radio set. In order to obtain room for the swing of the propeller, the car was necessarily placed at some distance below the hull; and the various controls and the access ladder between the car and the hull are contained within large streamlined casings.

ACCOMMODATIONS FOR OFFICERS AND CREW.

The accommodations for the officers and crew of the *ZR-3* are along the keel, as in the *Shenandoah*, but are considerably more elaborate, and comprise: (*a*) a cabin for the commanding officer, with bed, table, chair, and cupboard; (*b*) two sleeping cabins for the officers, each with two beds and two chairs; (*c*) an officers' wardroom with a table and seats; (*d*) six sleeping cabins for the crew, each containing two beds and two seats; (*e*) two mess-rooms for the crew, each with a table and seats; (*f*) two toilets with lavatory and water-closet fittings.

PROVISIONS FOR CARGO.

The keel contains 20 compartments for carrying passengers' baggage, mails, or other light cargo. Locked compartments for provisions, drinking water, and spare parts are also provided in the keel (Figure 19).

POWER CARS.

The five power cars are of stream-line form, with cross-sections much more nearly rectangular than in the cars of the *Shenandoah*. The cars and the engine foundations are constructed of duralumin girders, rolled sections, and plating. The thrust and propeller bearings are carried on an extension of the engine foundations.

As in the *Shenandoah*, the radiators are built into the forward end of the cars, and the flow of air is controlled by shutters.

The cars are suspended from the hull by stream-line duralumin tubes and steel wires. Access to the cars from the hull is by folding ladders.

Figures 20 to 24 show the power cars and engines of the *ZR-1* and *ZR-3*.

THE POWER PLANT.

The *ZR-3* is fitted with five 400 H. P. Maybach engines of a radically new design. The engines are placed in separate cars, one on the center line aft, and two pairs winged out upon the sides (Figures 19 and 20). The *Shenandoah* now has five Packard 250-300 H. P. engines disposed in a similar manner, although there was originally a sixth engine immediately behind the control compartment in the forward car.

During the war the Maybach six-cylinder vertical engine became famous for its reliability and economy, and was widely thought to be the ideal airship engine. For the *Shenandoah*, the Packard Company designed a special six-cylinder vertical engine which in reliability and performance has surpassed even the famous Maybach six. These Packard sixes were installed in place of the Liberty twelves originally contemplated.

In view of the widespread belief in the virtues of the straight six-cylinder engine for airship purposes, and the specialization of the Maybach Company upon that type, it is very notable that in designing the engines for the *ZR-3*, the straight six

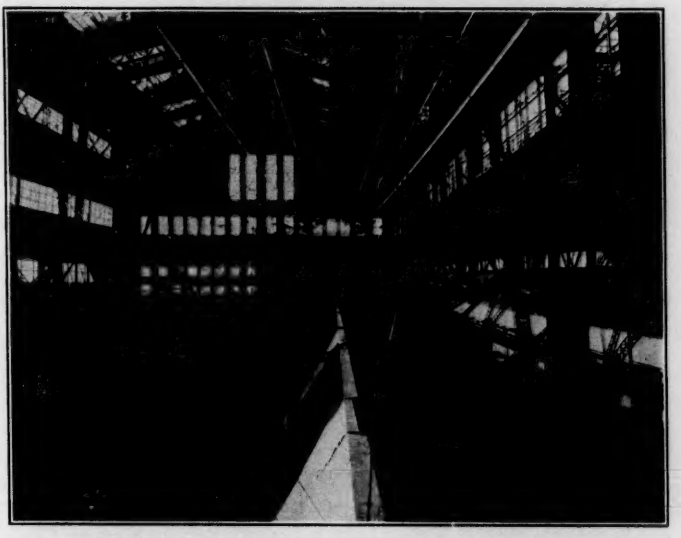


FIG. 1.—"SHENANDOAH" DURING CONSTRUCTION. VIEW ALONG TOP OF KEEL.

FIG. 2.—"ZR-3." CELL NO. 8 INFLATED FOR TEST. IN UPPER CENTER IS SHOWN AUTOMATIC GAS VALVE FOR RELIEVING EXCESS PRESSURE. NOTE THE FIVE-SIDED KEEL.

FIG. 3.—"ZR-3." OUTER COVER LACED TO HULL. NOTE FLAT TOP OF KEEL, EQUALIZING RING (UPPER CENTER) AND EXHAUST TRUNK FRAMING.

FIG 4.—"SHENANDOAH." SHOWING TRIANGULAR SHAPED KEEL STRUCTURE AND BOX GIRDERS FROM WHICH FUEL TANKS, BALLAST BAGS, ETC., ARE SUSPENDED.

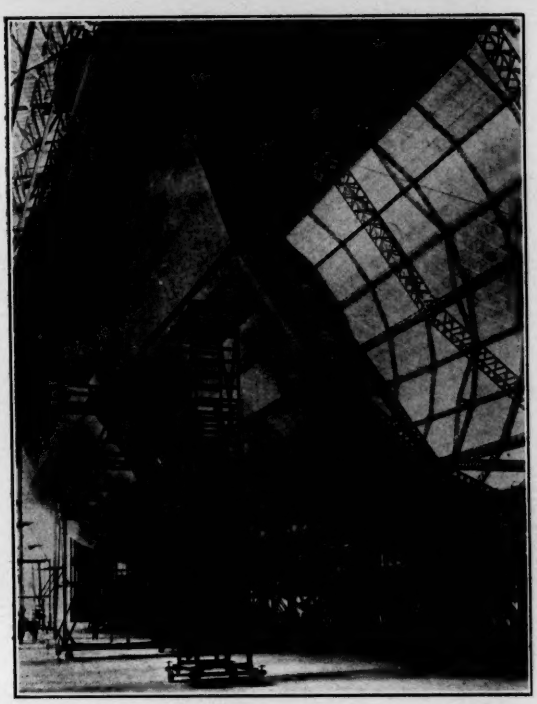


FIG. 5.—"ZR-3." OUTER COVER. TYPES OF STAGING USED TO WORK ON COVER ARE SHOWN, AND DIAMOND SHAPED ELEMENTS OF MAIN TRANSVERSE FRAMES.

FIG. 6.—"SHENANDOAH." MAIN FRAME CONSTRUCTED ON THE FLOOR AND HOISTED INTO POSITION. SHOWS KEEL AND ELEMENTS OF THE TRANSVERSE FRAME.

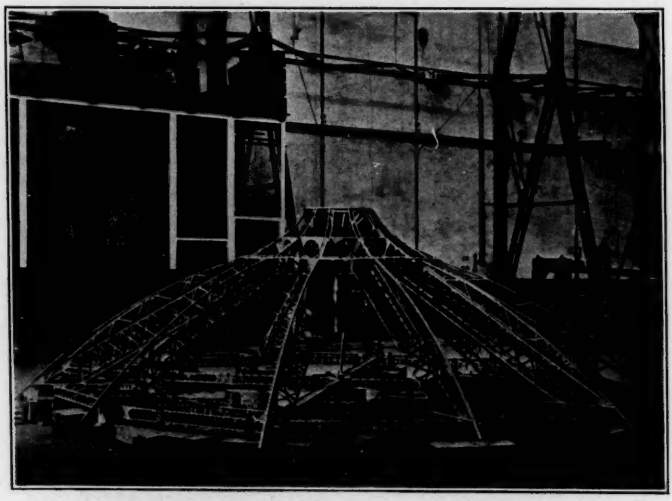


FIG. 7.—"ZR-3." BOW STRUCTURE, SHOWING REVERSE CURVE TO SUPPORT MOORING SPINDLE.

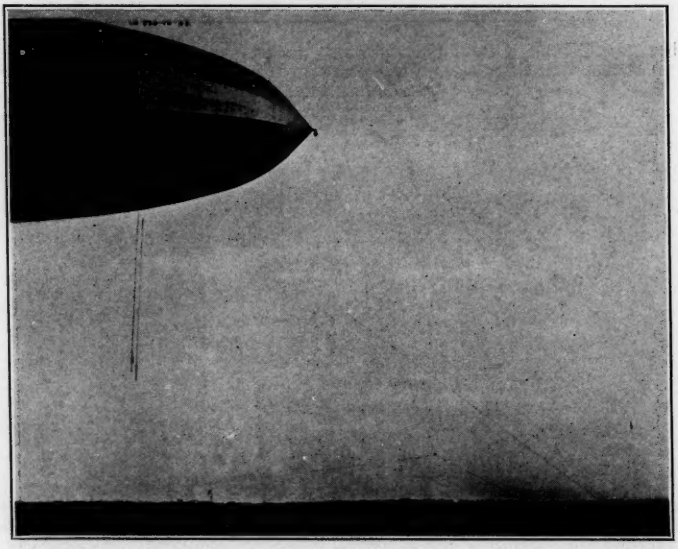


FIG. 8.—"SHENANDOAH" BEING LANDED BY TRACTORS. NOTE DIFFERENCE FROM BOW OF "ZR-3" SHOWN IN FIG. 7.

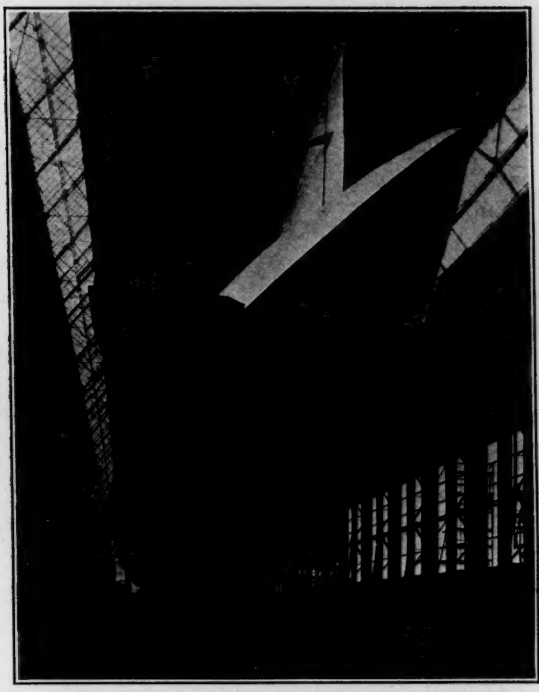


FIG. 9.—"ZR-3." SHOWING CHANGE OF FORM IN WAKE OF FINS, ALSO HORIZONTAL AND VERTICAL RUDDERS.

FIG. 10.—"ZR-3." FORWARD PART OF CONTROL AND PASSENGER CAR. NOTE NEW TYPE OF BOX GIRDERS.

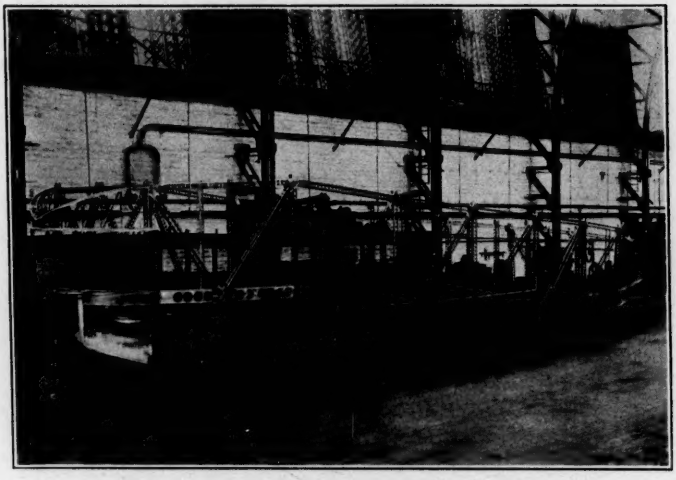


FIG. 11.—"ZR-3." SIDE VIEW OF CONTROL AND PASSENGER CAR, SHOWING FRAME WORK.

FIG. 12.—"ZR-3." VIEW FROM ABOVE CONTROL AND PASSENGER CAR, LOOKING FORWARD. NOTE BRACING.

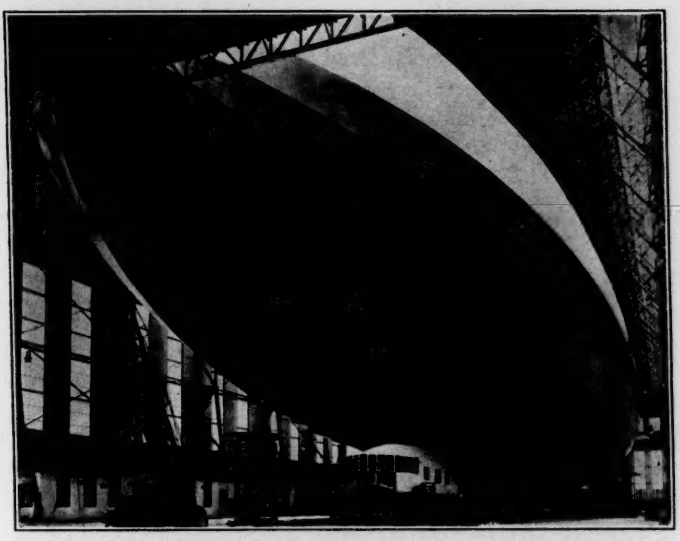


FIG. 13.—"ZR-3." NEAR COMPLETION AT FRIEDRICHSHAFEN. NOTE LARGE WINDOWS IN CONTROL AND PASSENGER CAR.

FIG. 14.—"ZR-3." VIEW ALONG CORRIDOR OF PASSENGER CAR, SHOWING DAYTIME ARRANGEMENTS.

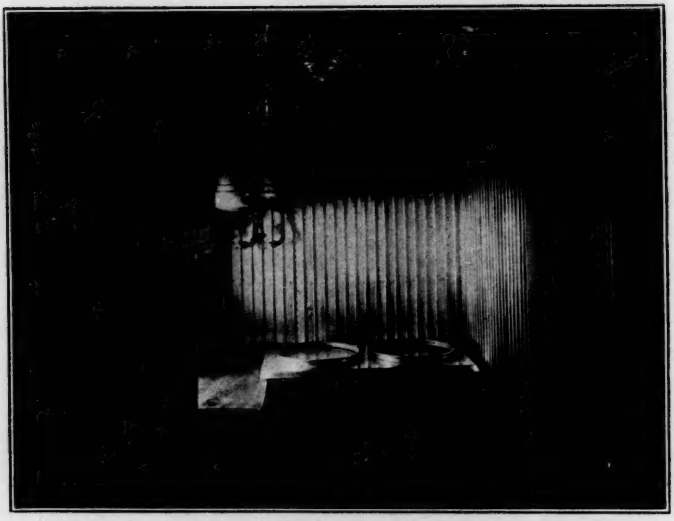


FIG. 15.—"ZR-3." KITCHEN.

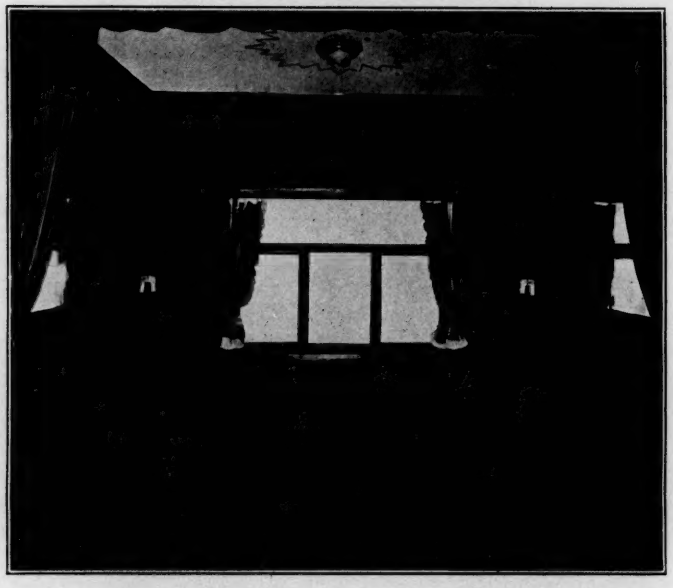


FIG. 16.—"ZR-3." ONE PASSENGER SECTION ARRANGED FOR DAYTIME USE. BACK OF SEATS LIFT UP TO FORM UPPER BERTHS AT NIGHT.

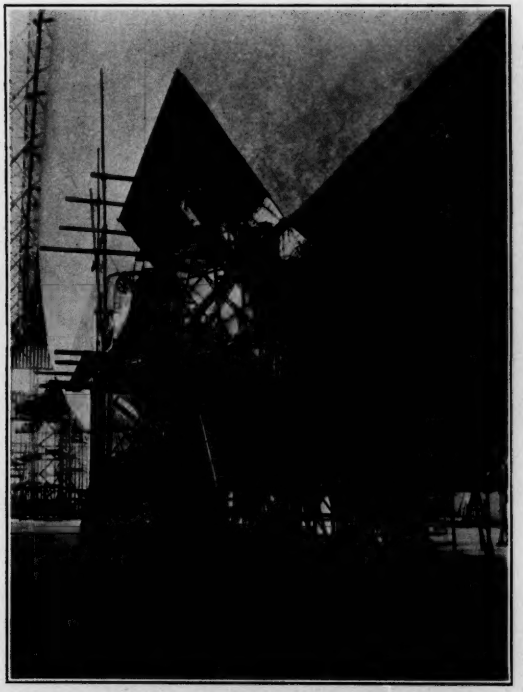


FIG. 20.—"ZR-3." SIDE POWER CAR.

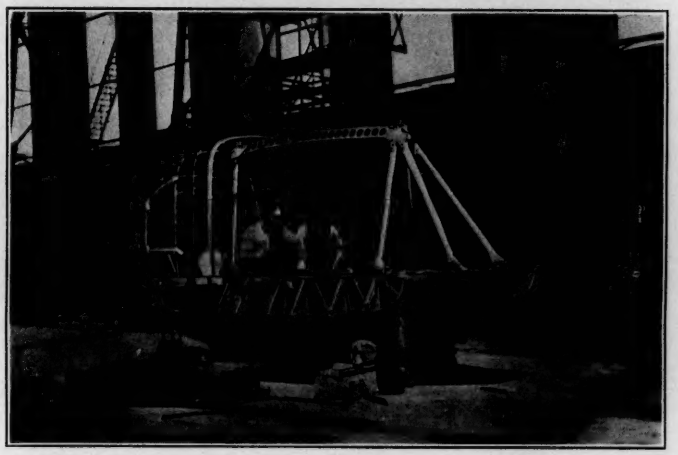


FIG. 21.—"ZR-3." POWER CAR UNDER CONSTRUCTION.

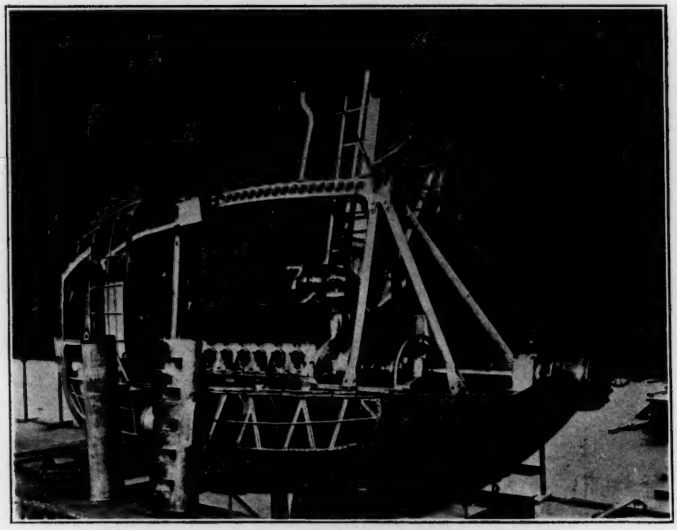


FIG. 22.—"ZR-3." SHOWING ENGINE INSTALLED IN POWER CAR, AND ACCESS TO HULL.

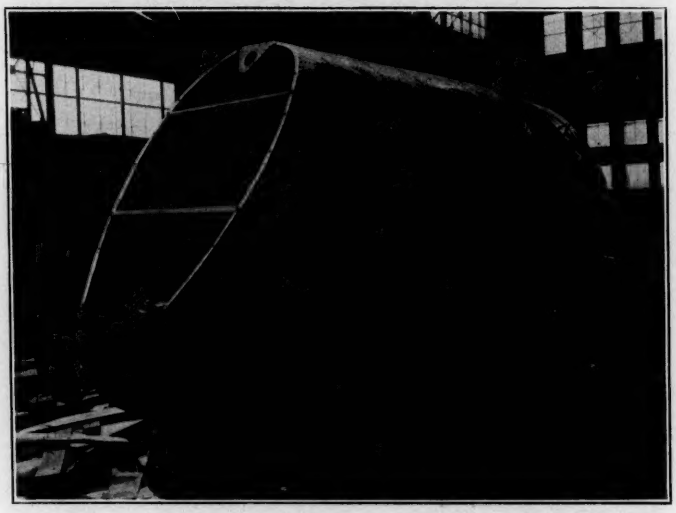


FIG. 23.—"SHENANDOAH." QUARTERING VIEW OF A POWER CAR.

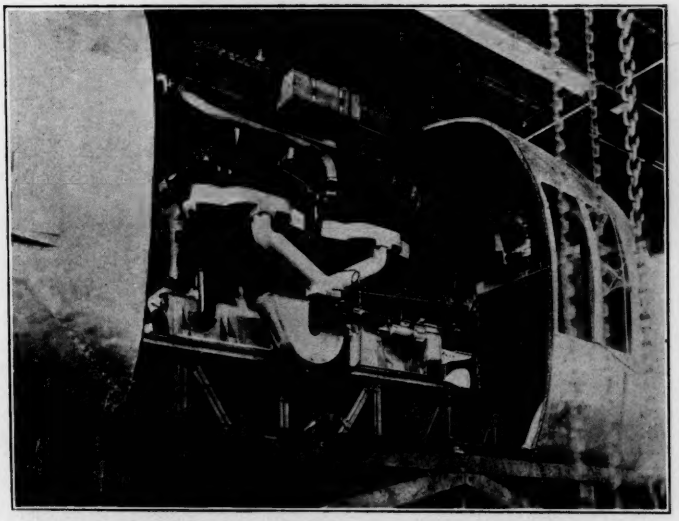


FIG. 24.—"SHENANDOAH." SHOWING ENGINE INSTALLED IN POWER CAR.

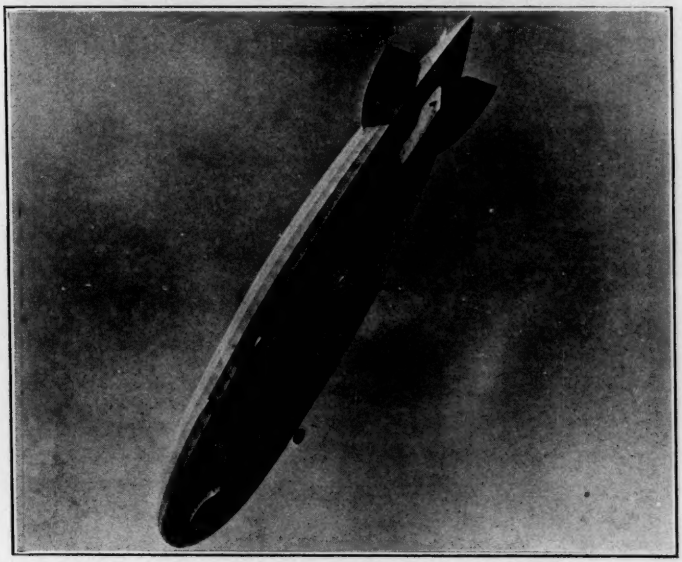


FIG. 25.—VIEW FROM UNDERNEATH "ZR-3" TAKEN UPON ARRIVAL IN THE UNITED STATES.

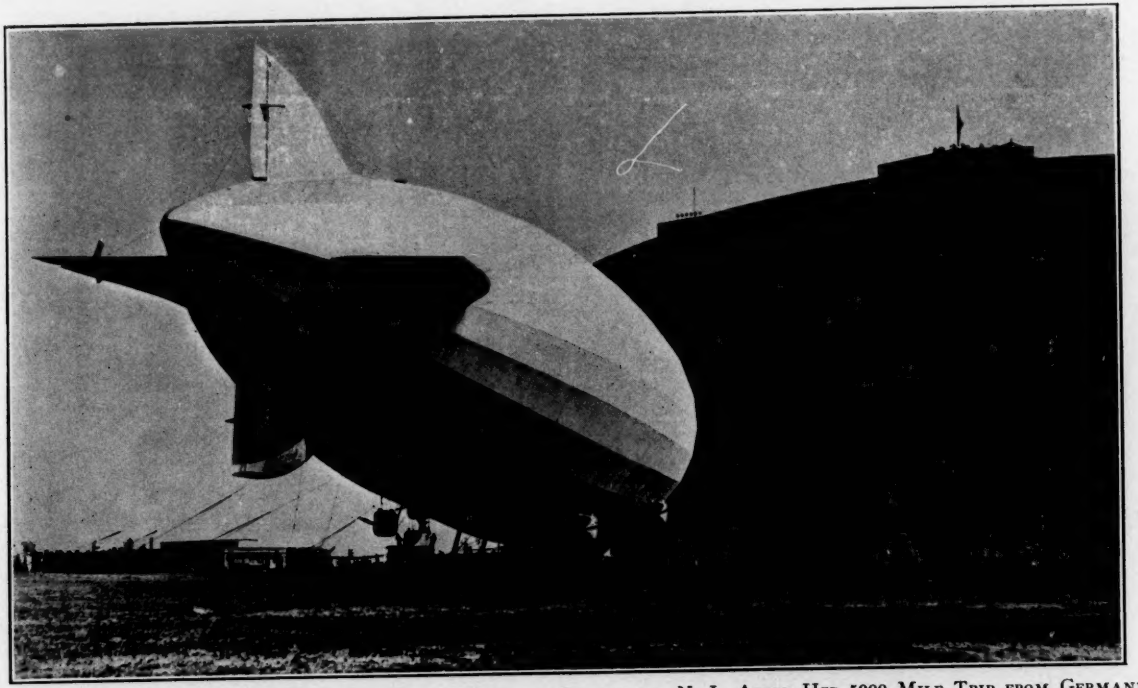


FIG. 26.—THE "ZR-3" BEING PLACED IN THE HANGAR AT LAKEHURST, N. J., AFTER HER 5000 MILE TRIP FROM GERMANY.

was discarded, and a wholly new design of 12-cylinder V engine adopted.

Each engine is designed to develop 400 brake horsepower at 1400 revolutions per minute. All five engines are reversible, and both starting and reversing are effected by compressed air, just as in marine Diesel engine practice. The propeller is coupled directly to the motor, except for an elastic coupling incorporated in the fly wheel, and a dog clutch.

In the *Shenandoah* none of the engines are reversible; but two of them are fitted with geared reversing propellers. Two other engines of the *Shenandoah* have direct driven non-reversing propellers, and the fifth engine has a geared non-reversing propeller.

The crank shaft of the new Maybach engine is completely housed in the upper part of the crank case. All the main bearings are of the roller type—a form held to give trustworthy service and practically unlimited life of the bearing. The cylinders, twelve in number, are arranged in V form. Between the two rows the cam shaft and valve operating gear are arranged. The cylinder blocks are made of grey cast iron with the water jackets welded on. The cylinder heads are so designed that equal cooling is given to all parts. Each cylinder is provided with one large inlet valve, two smaller exhaust valves, and two sparking plugs. As usual, the valves are operated by means of push rods. The cam shaft is driven by gearing from the crank shaft, the same drive serving to operate the auxiliaries. Four carbureters of the latest Maybach pattern are arranged between the two rows of cylinders. The exhaust pipes are carried along the outside of each row of cylinders.

Each motor is equipped with two twelve-cylinder Bosch magnetos, driven through an elastic coupling in order to avoid vibration. The ignition apparatus is arranged to work in either direction of rotation without alteration. Control of the ignition is secured by rotation of the driving shaft, the arrangement being such that the intensity of the spark is the same

for all positions. The circulating water pump delivers equal volumes of water for both directions of rotation. Lubrication is effected by means of one circulating and two suction pumps, the drives to which are such that the direction of working of the pumps is not altered by the reversal of the motor. The oil is withdrawn from the casing by the suction pumps at two points, and is then circulated through a filter to the main and other bearings.

When the engine is being started, air from a receiver is admitted through a distribution system, and caused to enter each cylinder at the correct instant. Starting valves, operated by the compressed air supply, are provided on each cylinder, and are shut down automatically as soon as the engine picks up on the ignition. It is claimed that although very little air is used, the torque at starting is large. The reversal of the direction of rotation is effected by sliding the cam shaft. A two-stage air compressor provides the starting air, and can be coupled to or uncoupled from the engine crank shaft while the engine is running. It is claimed that it is impossible to start the engine in a wrong manner, as the starting levers, the throttle, and reversing levers, and the ignition control are interlocked in such a way that they can be operated only in the correct direction. The interlocking arrangements render it impossible to start the engine otherwise than in the direction fixed by the position of the cam shaft. In addition, they are such that the ignition control is always set correctly for the direction in which the engine is to run. It is, moreover, impossible to reverse the engine if the throttle lever is not in the neutral position, and before the throttle lever can be moved the reversing lever must be in one or other of its end positions. Over and above these safety arrangements, the engine is fitted with an automatic device, whereby it is brought to rest immediately the lubricating oil pressure falls below a given value.

The following table gives the principal data on the motors:

MAYBACH REVERSIBLE ENGINE "VL-1."

Maximum brake horsepower	400
Maximum R. P. M.	1400
No. of cylinders, in V form	12
Estimated weight for service condition, approximately, pounds per H.P.	4.84
Total weight empty, pounds	1870
Total weight service condition, pounds	1935

(The above weights include the starter, reversing gear, air flasks and compressors.)

Gasoline consumption at 400 H. P. is .462 pound per horsepower hour, reducing to .410 pound per horsepower hour at about two-thirds full power. Most economical power can be adjusted within limits. Oil consumption 4.4 to 6.6 pounds per hour.

Bore, inches	5.5
Stroke, inches	7.1
Piston displacement per cylinder, cubic inches	169
Piston speed, feet per second	27.6
B. H. P. per cylinder	33.4
Compression ratio	5.8
Mean effective pressure, pounds per square inch	109
Mechanical efficiency (approximately), per cent	82
Thermal efficiency (approximately), per cent	36
Diameter of inlet valves, inches	2.6
Lift of inlet valves, inch	0.53
Number of inlet valves per cylinder	1
Diameter of exhaust valves, inches	1.81
Lift of exhaust valves, inch	0.39
Number of exhaust valves per cylinder	2

Some troubles developed in the trials, as was only to be expected with such a radically new design of engines. The chief difficulties experienced were in the balancing of the engine, cracking of the original crank cases, heating of the thrust bearings, and carbureter troubles.

The *ZR-3* arrived at Lakehurst, N. J., on October 15, 1924, having made the trip from Friedrichshaven, Germany, in approximately 81 hours. She was commanded by Dr. Hugo

Eckner on this trip and operated by a German crew. The distance covered was 5066 miles at an average speed of 62.35 miles per hour.

Lieut. Commander Sydney M. Kraus, U. S. Navy, who acted as inspector of the *ZR-3* while building and made the trip on board across the Atlantic, stated that the only casualty on the trip was the breaking of a stud of a rocker arm of one motor.

The Secretary of the Navy announced on October 16, at a luncheon to the officers and crew that brought the *ZR-3* to this country, that this big airship would be renamed *Los Angeles*.—EDITOR.

SOME EFFECTS OF FUEL MEASURING PUMP DESIGN ON DIESEL ENGINE PERFORMANCE.

By Lieut. (J.G.) J. E. Canoose, U. S. Navy, Member.
Navy Yard, New York.

Excessive breakage of pistons and cylinder heads caused by burned exhaust valves falling off their stems has been experienced with the R-21 to 27 class submarine engines in service. Hoping to determine the cause of the exhaust valve failures, a study of the design was made.

A study of the routine repairs made in the division of six boats over a period of a year showed very plainly that the boat having the best performance did a disproportionate amount of repair and adjustment to the fuel measuring pumps. Accordingly, a detailed study of the fuel measuring pump design was begun.

While the study was begun on the R-21 to 27 class engines, it was extended to the S-48 to 51 and V-1 to 3 classes. The fuel measuring pump design of these three classes of engines is essentially the same.

The types and horsepowers of these engines are:

R-21 to 27 class—4 cycle— 500 B.H.P.
S-48 to 51 class—2 cycle— 950 B.H.P.
V-1 to 3 class—2 cycle—2250 B.H.P.

A major part of the troubles existent in the engines in service as listed below are believed due to faulty fuel distribution system design.

4-CYCLE, 500 H.P., R-21 TO 27 CLASS.

(*a*) Excessive exhaust smoke at cruising speeds.
(*b*) Exhaust valve failures.
(*c*) Exhaust valve cage failures.
(*d*) Cylinder head failures due to heat cracks around exhaust cage seats.
(*e*) Cylinder head breakage, occurring when an exhaust valve disc breaks off while engine is running; and is driven through the head by the piston.
(*f*) Piston breakage, due to same cause as head breakage in (*e*).
(*g*) Exhaust header failures due to expansion resulting from afterburning.
(*h*) Engine foundation failures due to power unbalance in various cylinders.
(*i*) Cylinder tie-rod troubles (usually believed foundation trouble and usually resulting in cracked camshaft housings).
(*j*) Vertical camshaft gear troubles.
(*k*) Motor armature explosions due to failures of overspeed governors to function quickly.

2-CYCLE, 950 H.P., S-48 TO 51 CLASS.

(*a*) Piston failures.
(*b*) Cylinder failures.
(*c*) Cylinder head failures.
(*d*) Wrist pin failures.

2-CYCLE, 2250 B.H.P., V-1 TO 3 CLASS.

These engines have not been run in service; but their block test data has been analyzed.

The characteristics of the fuel measuring pump should include:

(*a*) Extreme simplicity of adjustment.
(*b*) High volumetric efficiency.
(*c*) It should be uniformly timed to all cylinders.
(*d*) A tappet adjustment should effect not more than one plunger, *i.e.*, there should be no group adjustments.
(*e*) Plunger packing should be designed so as to be independent of the personal element of assembly and adjustment.
(*f*) Vacuum should not be generated within plunger chambers of the pump at any time, *i.e.*, there should be no pressure reversals across the plunger packing.
(*g*) Valves should have minimum lift to minimize inertia tossing.
(*h*) Cyclic speeds must be the minimum.
(*i*) Plungers should be floated in their driving member.

The spray valve should have the functional characteristics required by the following discussion:

Referring to Figure 1 which is the lower part of a spray valve, 4-cycle type—

The annular space "*a*" is constantly filled with spray air; the pressure (usually about 450 pounds above compression pressure) blows oil to the cylinder when needle opens. The atomizer rings "*b*" entrain as much oil as their surfaces and perforations will hold, this capacity depending on: (*a*) the time interval, *i. e.*, the time between the delivery of oil to the valve and the opening of the valve; and (*b*) viscosity, adhesion, cohesion, volatility, etc., of the oil at the working temperature of valve. The fuel needle (closed) traps any oil at "*c*" that passes the atomizer rings, preventing it from going to the cylinder prematurely. The temperature of the above spray valve parts depends on the temperature of spray air delivered from cooler and on temperature of cylinder head as a whole; but rarely exceeds 150 degrees F.* Assuming a spray air

*These figures were used by Mr. W. S. Burn, M. Sc., Graduate, in his paper on "Diesel Engine Flexibility" read before the North East Coast Institution of Engineers and Shipbuilders. They were used by him only to demonstrate effect and are so used here However, substitution in the equation shows that a few degrees difference in T_1 has little effect on T_2.

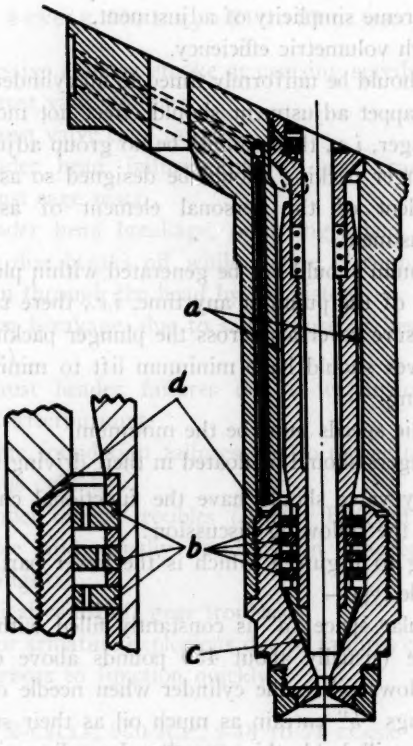


FIGURE 1.

temperature of 150 degrees F. at say 1000 pounds pressure, it is readily seen that when this air is expanded to compression pressure it will have a decided refrigerating effect within the cylinder—(For initial T. of 150 degrees)—assuming 500 pounds compression.

$$T_2=611\left(\frac{515}{1015}\right)^{\frac{.4}{1.04}}=503 \text{ degrees F. absolute } 42 \text{ degrees F.}$$

This pocket of injected air must be heated from heat within

the cylinder. Most designers have tried to meet this condition by use of a spray valve which allows a very small amount of oil to collect around the seat of fuel needle, which goes to the cylinder unatomized ahead of the air charge. This fuel burning around the spray orifice supplies the heat needed to raise spray air injected to cylinder temperature. In case of a cylinder misfiring, the following obtains: Assume a compression of 500 pounds and resulting temperature of 1170 degrees F., or 1630 degrees F. absolute; and assuming injection air equals 1/12* of the compressed cylinder air then we should have

$$\frac{12 \times 1630 \text{ plus } 1 \times 503}{12 \text{ plus } 1} = 1543 \text{ F. absolute, } 1630 - 1543 = 87$$

degrees temperature drop for the entire cylinder charge. Now, assuming that there is no oil collected around the valve needle, and that 1/12 of the injection angle is wasted before oil and air mixture begins to blow through, it is seen that about 1/144 part of the total charge within the cylinder (pocketed around the spray nozzle) is at its admission temperature, 42 degrees F. plus such heat as it has received by conduction and radiation from the cylinder charge. The absolute need for a small portion of unatomized "pilot oil" at the beginning of the stroke is obvious.

The number of atomizer plates to be used depends in general on: (*a*) The engine load. (*b*) Engine mean R.P.M. (*c*) Engine temperature, and (*d*) Oil viscosity. The number of plates should be such that they will entrain the full charge of oil except a very small amount (about 5 per cent of the total) which should run down around the needle seat ready to enter as a "pilot oil" charge to generate the heat absorbed by the spray air as it enters.

*These figures were used by Mr. W. S. Burn, M. Sc., Graduate, in his paper on "Diesel Engine Flexibility" read before the North East Coast Institution of Engineers and Shipbuilders. They were used by him only to demonstrate effect and are so used here. However, substitution in the equation shows that a few degrees difference in T_1 has little effect on T_2.

(*a*) In reference to the engine load, it should be noticed; that what is a correct number of plates for full load, should seem to be too many for partial load, because the smaller amounts of oil used at light loads should be entrained by the first plate, and thus delayed in getting through the lower plates. But at full powers a greater amount of pilot oil is needed than at reduced powers, when the spray air pressure being lower, less refrigeration is taking place during expansion of spray air to cylinder pressure. It is thus seen that the variable load effect is not so adverse as it at first seems.

(*b*) The effect of speed variation is different to that of load variation. Underway, with engine on the screw, the maximum speed is coincident with the maximum fuel consumption, but while charging, the speed can be kept constant (with adequate generator field regulation) at all loads, or nearly so. In that, the work on the screw is most important, it will be considered. The only effect speed has on the functioning of the spray valve is, that with all other conditions constant, increased speed allows less time for the fuel to run through the atomizer plates. The higher spray air pressure used at higher speeds does accelerate the injection, but does not affect the passage of oil down through the atomizer plates before the valve opens. Therefore, it does not affect the passage of oil past the plates to form pilot oil. So at low speeds, there is a greater time interval for the passage of oil by the plates. This compensates partially for the adverse effects of the smaller amounts of oil measured at the lower speeds.

(*c*) The temperature of the engine is generally within the operator's control, thus it may be kept nearly constant.

(*d*) The viscosity of the oil, though not under the operator's control, can be met by him in changing the number of atomizer plates to suit.

Regarding the urgent need of pilot fuel, reference is made to the experience European engineers are having with engines burning tar oils. Tar oils have a spontaneous combustion

temperature nearly double that of Diesel oils. As engines will not start satisfactorily on this oil, an additional "pilot pump" is provided to deliver a very small quantity (5 to 10 per cent of the total fuel) of regular Diesel oil or kerosene, ahead of the charge of tar oil for ignition purposes.* When the engine is running at, or nearly, full load, the pilot pump is cut out, but when the load is small, the engine misses, and it is necessary to use the pilot pump. It is found that the engine will run at lower loads when both pumps are run on tar oil than when only the main pump is running. This is due to the fact that the pilot oil positively enters the cylinder unatomized ahead of the spray air, thus anticipating the drop in temperature around the spray nozzle. The important result of staggering the holes in the atomizer plates (*b*), Figure 1, is that a vigorous turbulence is set up between the plates, thus sweeping them clear of oil and distributing the oil through the entire arc of admission at each opening of the valve. The pulverizing of the oil in the passing through this labyrinth of annuli is incidentally accomplished. This distributing or "braking action" of the atomizer elements is very essential to obtain proper progressive combustion, and to avoid undesirable detonation.

A leaking spray valve, besides losing some of its charge of oil, is upset in its desired functioning; because the slow passage of air through the valve sweeps out any pilot oil accumulated around the seat. The necessity for pilot oil in the type of combustion chamber shown at (*b*), Figure 2, is greater than in that shown at (*a*) of same figure. In the former type of combustion chamber the oil and air has a much shorter distance to travel to strike the piston after leaving the spray valve nozzle, than in the latter type. In the former case any oil carried in this expanding air may by virtue of its greater

*Reference: (*a*) Fuel and lubricating oils for Diesel Engines by W. Schenker, Chief Engineer of Sulzer Freres, Switzerland. (*b*) Paper by Harold Moore, M.Sc., Tech., read before the North East Coast Institution of Engineers and Shipbuilders, entitled "Some Factors Affecting the Utilization of Heavy Fuel Oils in Internal Combustion Engines."

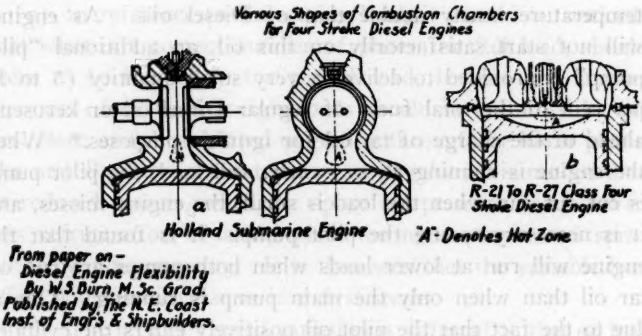


FIGURE 2.

inertia be carried to the piston, before it is heated sufficiently to fire. By the time the spray air has carried down the large quantities of oil, and sufficient turbulence of air has been set up in the cylinder to transfer sufficient heat for ignition to the charge of oil, the piston will have started on its downward stroke, thus decreasing the sensible heat appreciably. Assume, as per calculation above, that the initial 1/12 of injection angle is wasted before oil is carried to the cylinder. This is roughly 3.5 degrees of crank angle, and in R-21 to 27 class corresponds to .014 inch of piston travel from dead center. This causes a reduced compression of 10 pounds and corresponding reduction in temperature of compression besides the rapid heat loss at this point to the circulating water. All of these factors, at best, contribute to delayed combustion with consequent afterburning or to total misfiring. In the above computation the 3 degrees or 4 degrees advance given the spray valve was not considered because this lead is usually given to compensate for lags incident to a spray valve functioning perfectly.

This covers the general functioning of the spray valve, assuming that the timing of fuel delivery *to the spray valve* is the same to all cylinders. But the latter factor is not even approximated in the R-21 to 27 class, 4-cycle engine. In fact, as the following explanation will show, the performance

in this particular factor is as far as possible from the ideal. Refer to Figure 3; this diagram shows the engine 4-stroke cycle in one circle, thus one semi-circle equals one revolution of the crankshaft. The cyclic degrees are marked in the rectangular enclosures from 0 degree to 720 degrees, also, in these the cylinder time sequences and firing sequences are shown. The rotation is shown by the arrow, the firing sequence running anti-clockwise, and time sequence running clockwise. The diagram shows the cycle of all cylinders successively super-imposed in accordance with their firing and time sequences. The firing sequences are also numbered radially, each number within the zone for that cylinder. Each zone shows the crank pin position at points of opening and closing of the spray valve, inlet valve and exhaust valves. All angles

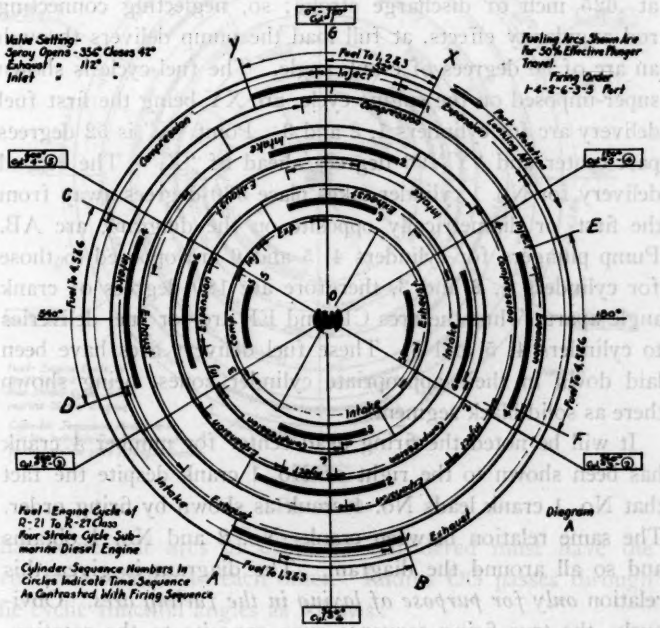


FIGURE 3.

of the diagram are drawn to scale so that two degrees of diagram equals one measured degree. Thus a cylinder whose crank angle is 120 degrees from its leader measures only 60 degrees. The firing order is 1-4-2-6-3-5 as shown.

Consider now the fuel pump cycle. The fuel pump runs at crankshaft speed; so each plunger delivers oil to its spray valve once each revolution. This gives two deliveries of fuel per cycle with deliveries 360 degrees apart. This on the 720 degrees diagram (Figure 3) places them diametrically opposite. The specified timing of the fuel pump is (for the U.S.S. R-24 port engine) for plungers 1, 2 and 3, to be at delivery dead center when No. 1 working crank is 52 degrees past top dead center on expansion stroke. The stroke of the pump is 1.25 inches, and at full load the suction valves close at .625 inch of discharge stroke; so, neglecting connecting rod angularity effects, at full load the pump delivers through an arc of 90 degrees of crank angle. The fuel cycle is shown super-imposed on the engine cycle, arc XY being the first fuel delivery arc for cylinders 1, 2 and 3. Point "X" is 52 degrees past center and "Y" 90 degrees ahead of "X." The second delivery for No. 1 cylinder takes place 360 degrees away from the first, or diametrically opposite on the diagram, arc AB. Pump plungers for cylinders 4, 5 and 6 are opposed to those for cylinders 1, 2 and 3, therefore are 180 degrees of crank angle apart. Thus the arcs CD and EF are for fuel deliveries to cylinders 4, 5 and 6. These fuel delivery arcs have been laid down in their appropriate cylinder zones, being shown there as solid black segments.

It will be noted the firing dead center for number 4 crank has been shown to the right of No. 1 crank despite the fact that No. 1 crank leads No. 4 crank as shown by firing order. The same relation between cranks No. 2 and No. 4 obtains and so all around the diagram. The diagram is given this relation *only for purpose of laying in the fueling arcs.* Obviously, the true firing sequence runs opposite to the rotation.

But consider cranks for cylinders No. 1 and No. 2. Crank No. 1 passes firing center 240 degrees ahead of No. 2, but the spray valves to cylinders No. 1 and No. 2 are fueled simultaneously. Therefore No. 2 is fueled, relatively, 240 degrees earlier than No. 1; so, in order to lay in the fueling arcs for No. 1 and No. 2 coincident as shown, No. 2 diagram must precede No. 1 by 240 degrees. As the fueling arcs are laid on the diagrams to indicate the time of fueling in relation to the cyclic time for any cylinder considered, it is obvious

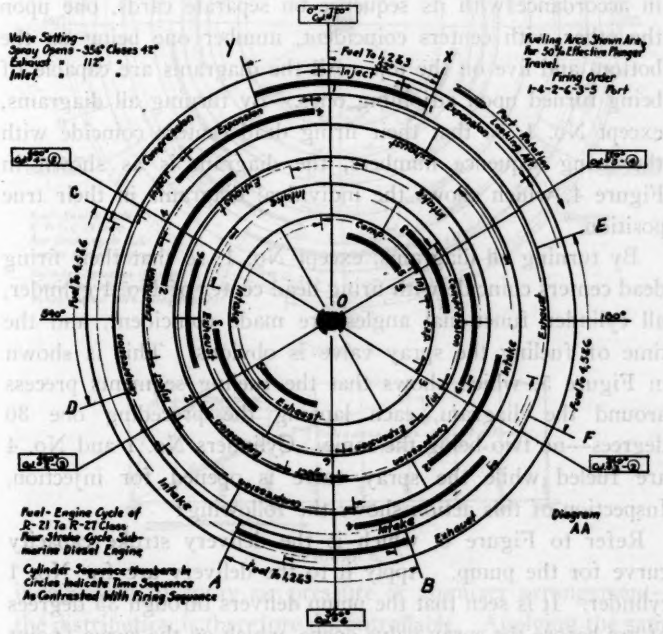


FIGURE 4.

that coincident arcs of cylinders considered must have the true time relation to each other. Radius OG passes through the cyclic function angles as follows:

Cylinder No. 1—injection.
Cylinder No. 4—compression.
Cylinder No. 2—intake.
Cylinder No. 6—intake—exhaust.
Cylinder No. 3—exhaust.
Cylinder No. 5—expansion.

the diagram in this position showing in relation to any one cylinder what is occurring in all other cylinders.

In the original diagram each cylinder diagram is mounted in accordance with its sequence on separate cards, one upon the other with centers coincident, number one being on the bottom and five on the top. All the diagrams are capable of being turned upon adjoining ones. By turning all diagrams, except No. 1, so that their firing dead centers coincide with the firing sequence numbers, the diagram is as shown in Figure 4, which shows the individual diagrams in their true position.

By turning all diagrams, except No. 1, so that their firing dead centers coincide with firing dead center of No. 1 cylinder, all cylinder functional angles are made coincident, and the time of fueling the spray valve is obvious. This is shown in Figure 5, which shows that the fueling segments precess around the diagram, each lapping the preceding one 30 degrees—no two being the same. Cylinders No. 1 and No. 4 are fueled while the spray valve is opened for injection. Inspection of this action shows the following:

Refer to Figure 6, which is the delivery stroke capacity curve for the pump. Apply it to the delivery arc for No. 1 cylinder. It is seen that the pump delivers through 33 degrees of arc before the spray valve opens, which on the curve (point X) is 53 per cent of the pump delivery. The pump delivery arc overlaps the injection angle at both ends. The injection angle is approximately 40 degrees. Referring again to Figure 6, it is seen that this 40 degrees of arc represents 35 per cent of the stroke. This oil is being distributed at a rate entirely

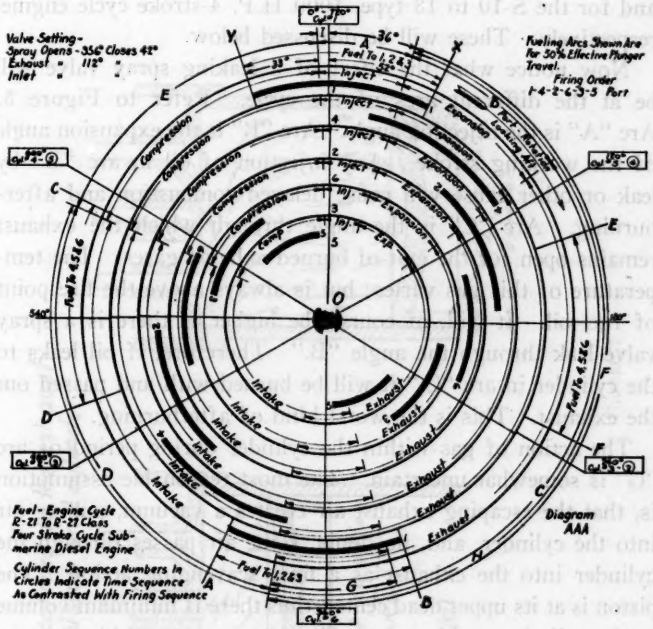

Valve Setting -
Spray Opens - 356° Closes 42°
Exhaust " 112° "
Inlet " 112° "
Fueling Arcs Shown Are for 50% Effective Plunger Travel.
Firing Order 1-4-2-6-3-5 Port
Fuel-Engine Cycle R-21 To R-27 Class Four Stroke Cycle Submarine Diesel Engine
Cylinder Sequence Numbers In Circles Indicate Time Sequence As Contrasted With Firing Sequence
Diagram AAA
Inject
Compression
Expansion
Intake
Exhaust

FIGURE 5.

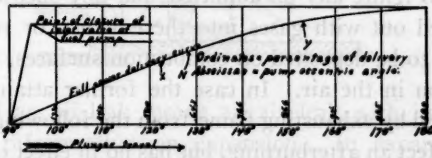


FIGURE 6.

independent of spray air pressure or atomizer arrangement—the distribution is therefore uncontrollable. Applying the same reasoning to cylinder No. 4, observe that it gets a large slug of oil (25 per cent) during the first half of the injection angle. This is likewise uncontrollable and obviously wrong.

Figures 8 and 10 with 9 and 11 show the engine-fuel pump cycles for the V-1 to 3 class 2250 H.P. 2-stroke cycle engine

and for the S-10 to 13 type, 1000 H.P. 4-stroke cycle engine, respectively. These will be discussed below.

Now notice what the effect of a leaking spray valve will be at the different arcs of the cycle. Refer to Figure 5. Arc "A" is the injection angle. Arc "B" is the expansion angle of the working stroke. Any injection of oil in arc "B" by leak or other cause will cause delayed combustion and afterburning. Arc "C" is the angle through which the exhaust remains open for the exit of burned exhaust gases. The temperature of this gas varies; but is always above the fire point of fuel oil. It will, of course, be higher if there is a spray valve leak through the angle "B." Therefore, if oil leaks to the cylinder in arc "C," it will be burned with and passed out the exhaust. This is the worst kind of afterburning.

The action of gas within the cylinder during period of arc "G" is somewhat uncertain. The most reasonable assumption is, that the escaping exhaust air creates a vacuum, pulling air into the cylinder, and, no doubt, some air passes through the cylinder into the exhaust as a final scavenging action. The piston is at its upper dead center, thus there is minimum volume in the cylinder at this point. The temperature within this arc is too low to ignite any oil admitted; but any such oil will be either carried out with gases into the exhaust, or retained in the cylinder to be deposited on combustion surfaces, or carried in suspension in the air. In case the former attains, the oil will be burned by exhausting flame from the following cylinder. This is in effect an afterburning, but has no ill effect on exhaust valves, except in causing accumulation of soot deposits. Arc "D" is the inlet angle after the closure of the exhaust valve. Any oil admitted to the cylinder during this arc will be hurled against the cylinder wall and piston by the incoming air, somewhat after the fashion shown in Figure 7. Of course, some of the oil will be carried in suspension in the air, and compressed with it, on the compression stroke. But it must be remembered that there is always spray air pressure in the

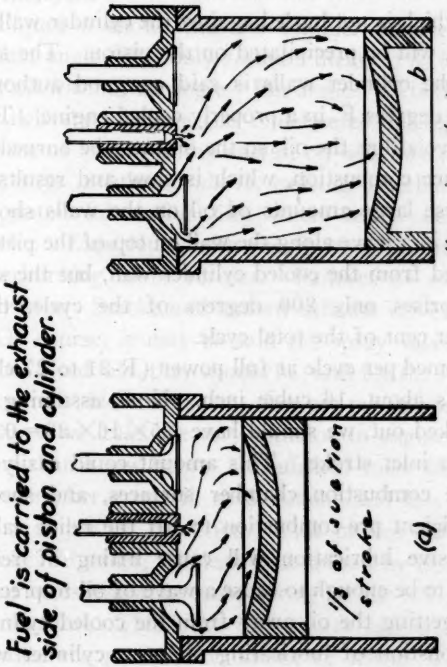


FIGURE 7.

valve, and an oil leak means an air leak with its attendant, excessive refrigeration on expansion. In expanding spray air to 500 pounds pressure, it was seen above, the temperature dropped to 42 degrees F. but on expanding to atmospheric, or lower pressure existent in the cylinder during the inlet stroke, the temperature will drop to

$$T_2 = 611\left(\frac{15}{1015}\right)^{\frac{.4}{1.04}} = 183 \text{ degrees absolute or } -278 \text{ degrees F.}$$

Of course the air leak is small, but so is the oil leak, so there is sufficient amount of air to chill the oil as it enters the cylinder. This is adverse to atomization, so little takes place—most of the oil, which is not hurled against the cylinder wall by the inrush of air, will be precipitated on the piston. The skin temperature of the cylinder walls is said on good authority to be around 275 degrees F. in a properly cooled engine. This is not enough to volatilize the oil, so the oil must be burned by progressive surface combustion, which is slow and results in smoke. Of course large amounts of oil on the walls should be driven upward in a wave along the wall on top of the piston, and thus separated from the cooled cylinder wall, but the suction stroke comprises only 200 degrees of the cycle, thus 200/720 or 28 per cent of the total cycle.

The fuel consumed per cycle at full power (R-21 to 27 class 4-cycle engine) is about .16 cubic inch. Now, assuming ¼ of the fuel is leaked out, we should have .25×.16×.28=.0112 cubic inch lost to inlet stroke. This amount could easily be entrained on the combustion chamber surfaces, and should hardly cause sufficient pre-combustion to lift the relief valve. Of course excessive lubrication will cause lifting of relief valves; but it has to be enough to cause a wave of oil to precede the piston, thus getting the oil away from the cooled cylinder wall. Skin combustion of lubricating oil on a cylinder wall will not cause pre-ignition.

Any oil leaking into the cylinder during angle "E" Figure 5 can only be entrained in the air and compressed with it. Possibilities for entrainment increase with increased temperature and pressure. Therefore, when we have least heat and pressure, the oil has greatest distance to fall to strike the piston. Any oil leaking in during this angle will undoubtedly cause pre-combustion. But with relief valves set at the specified 750 pounds (as in the case of R-21 to 27 engines) lifting pressure, considerable added heat is required to lift the valve.

The angle in this case is 134 degrees, or $\frac{134}{720}$ =18½ per cent of the cycle.

Figure 5 shows that no cylinder receives all of its fuel within angle "E," only four of them, 1, 3, 5 and 6, receiving any oil within that angle. But angle "E" is the only safe angle for delivery. Where fuel is delivered during any other angle, in case of leaking spray valves, oil will surely be lost in one of the ways mentioned above, causing afterburning, or dilution of piston lubrication film where leaks occur during intake stroke. Experience with the R-21 to 27 engines proves the afterburning to be serious. Failures of exhaust valves are excessive. Smoke and burned exhaust valves go together.

Of course, it may be said the spray valves should be kept tight. No doubt, they should be; and for the speed runs, acceptance trials, block tests, etc., they are put in good condition. But spray valve leaks are very common to engines which have run more than a few hours. Insufficient roller clearance in a spray valve rocker arm will cause smoke and loss of power with a valve in perfect condition. This proves that leaks will cause smoke. Further, this leak thus caused by insufficient clearance must be excessive to cause lifting of the relief valve from precombustion. This latter fact is very important to the operator because the pre-ignition knock or lifting of the relief valve is the only conclusive evidence he has, of a leaking valve, while the engine is in operation. Misfires in this type of engine are very commonly corrected by repairing leaky spray valves. On long trips spray valves are bound to develop leaks. Small leaks show no evidence of themselves except the smoke of the exhaust. They very rarely lift the relief valves; but the practice of cutting down spray air pressure to one cylinder to keep the cylinder firing is very common. It is obvious that a scored valve will leak throughout the entire cycle. Because of the greater difference in pressure, it will leak much faster during exhaust and inlet strokes than it does on compression and

working strokes. Therefore, when we have a leak large enough to cause pre-combustion, we know the greater percentage of the oil has gone out with the exhaust as smoke. Cutting down spray air to individual cylinders to prevent knocking or missing simply reduces the mean pressure difference carrying the oil through, thus making the leak rate lower.

Instruction books, in enumerating causes of smoke and misfiring, omit the spray valve leak.

Reference to Figure 3 will show that with a pump timed to deliver all fuel in angle "E," a leak to lift a relief valve or cause a knock of a certain force will leak oil only

$$\frac{133 \text{ degrees}}{679 \text{ degrees}} = \frac{\text{Compression angle}}{720 \text{ degrees} - \text{injection angle}} = 19.6 \text{ per cent}$$

as long as it now does in the case of No. 1 and No. 4 cylinders. For other cylinders the percentage scales upward according to the fuel arc position. When knocks occur in cylinders 1 and 4, about 80 per cent of the fuel leaking through is doing no work; and that causing the knocks is doing negative work.

Referring to Figures 8 and 9, it is seen that these diagrams being for a two-cycle engine are 360-degree diagrams. However, all the valving action angles are similar. The spray valves are fueled so that pairs 1 and 4, 2 and 5, and 3 and 6 are fueled alike, the pairs falling 120 degrees apart. Valves 2 and 5 are fueled during the injection angle. This causes fuel to be delivered to the cylinder in proportion to delivery curve, Figure 6. Timing for Figure 8 was taken from one of the V-1 engines. Timing of the fuel pumps on other V-boat engines varies. Now suppose the timing happens to so fall that two of the spray valves begin to receive oil during the last half of the injection angle. Referring to Figure 6, it is seen that 34 per cent of a charge should be delivered. This should, no doubt, cause a sudden rise in pressure with well regulated combustion. But the indicator cards for these

engines show combustion to be too erratic to recognize with certainty any irregularity of fuel distribution during injection; therefore nearly any rational analysis fails.

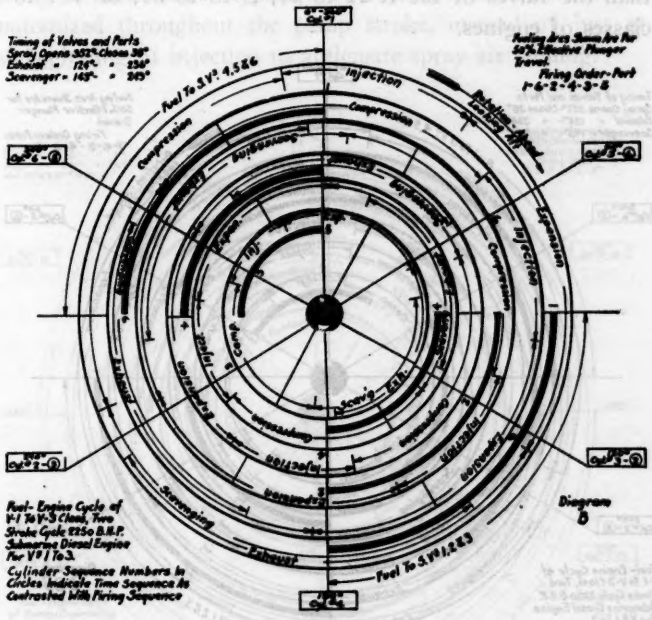

Timing of Valves and Ports
Spray Opens 352°
Closes 28°
Exhaust 176° 234°
Scavenger 169° 245°
Fuel To 5 V1 4 5 & 6
Injection
Compression
Scavenging
Exhaust
Expansion
Fuel To 5 V1 4 5 & 3
Diagram
Post-Engine Cycle of V-1 To V-3 Class, Two Stroke Cycle 2250 B.H.P. Submarine Diesel Engine For V1 To 3.
Cylinder Sequence Numbers in Circles Indicate Time Sequence As Contrasted With Firing Sequence

FIGURE 8.

While accurate information as to timed pump positions is not available, the cycle for the engines of the S-48 to 51 class of boats is identical to that of the V-1 to 3. Therefore, as neither class have timed pumps, the average cycle should be the same for both engines.

Figures 10 and 11 show the fuel engine cycle diagrams for the Bureau type 1000 H.P. engine. As will be noted, when the individual diagrams are all turned, so that their firing centers coincide, the fueling time for spray valves falls in pairs Nos. 1 and 6, Nos. 2 and 5, and Nos. 3 and 4. These

all fall at points within exhaust angles, and should cause afterburning if spray valves leaked. But due to different design characteristics, the Bureau type valve develops leaks less easily than the valves of the R-21 to 27, S-48 to 51, or V-1 to 3 classes of engines.

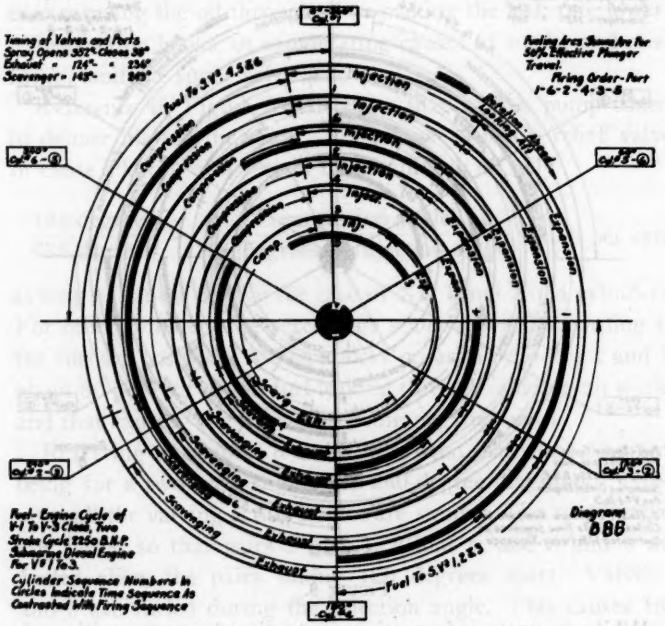

Timing of Valves and Ports
Spray Opens 352°- Closes 38°
Exhaust - 124°- - 234°
Scavenger - 145°- - 345°
Fueling Area Shown Are for 50% Effective Plunger Travel.
Firing Order - Port
1-6-2-4-3-5
Injection
Compression
Expansion
Exhaust
Scavenging
Fuel To 3 V's 4,5&6
Fuel To 3 V's 1,2&3
Fuel - Engine Cycle of V-1 To V-3 Class, Two Stroke Cycle 2250 B.H.P. Submarine Diesel Engine For V's 1 To 3.
Cylinder Sequence Numbers in Circles Indicate Time Sequence As Contrasted With Firing Sequence
Diagram 888

FIGURE 9.

Attempts were made to use pilot oil passages by passing the atomizer elements in the V-1 to 3 type engines, the aim being to supply pilot oil. The failure of this attempt was partly due to fueling some valves while open. In this case, an unatomized stream of oil is injected into the cylinder from the pilot port. Because, with the spray valve lifted, the pressure above the atomizer should be about the normal spray air pressure, but at the tip the pressure should be somewhat reduced. This pres-

sure difference should cause a change in the flow ratio of pilot oil to oil flowing over the atomizer elements during pump delivery strokes. Thus, the pilot porting should function merely as an atomizer bypass. Therefore, it should admit oil unatomized throughout the pump stroke, instead of just at the beginning of injection to anticipate spray air chilling.

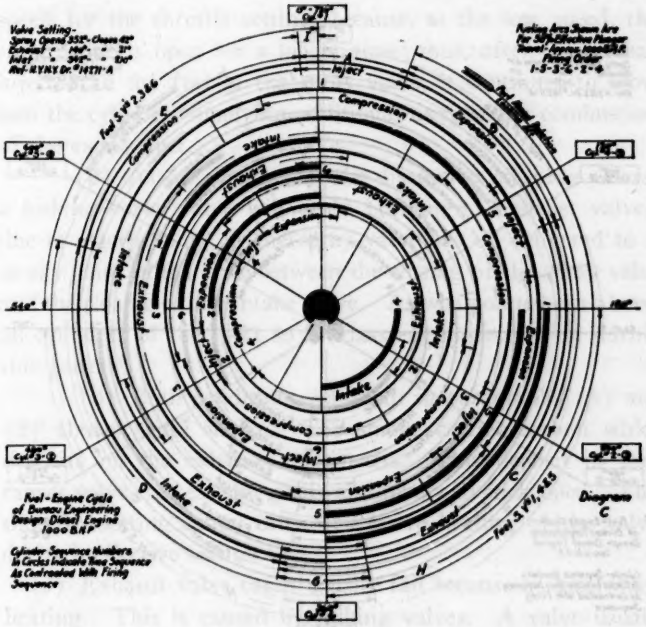


FIGURE 10.

Discussion of troubles of the R-21 to 27, 4-cycle engine as set forth above follows:

(*a*) The excessive smoking of these engines is common knowledge. It is partially due to overloading of individual or groups of cylinders at times when the other cylinders are underloaded. This is done to maintain cruising speeds. Fur-

ther, as set forth above, a leaking spray valve cannot do other than cause smoke. Leaking spray valves, leaking pump packing glands, leaking pump valves and unbalanced adjustments are the visible causes of underloading of individual cylinders as set forth above. High cyclic pump speeds, bad valve design and vacuum generation in plunger chambers are the invisible causes.

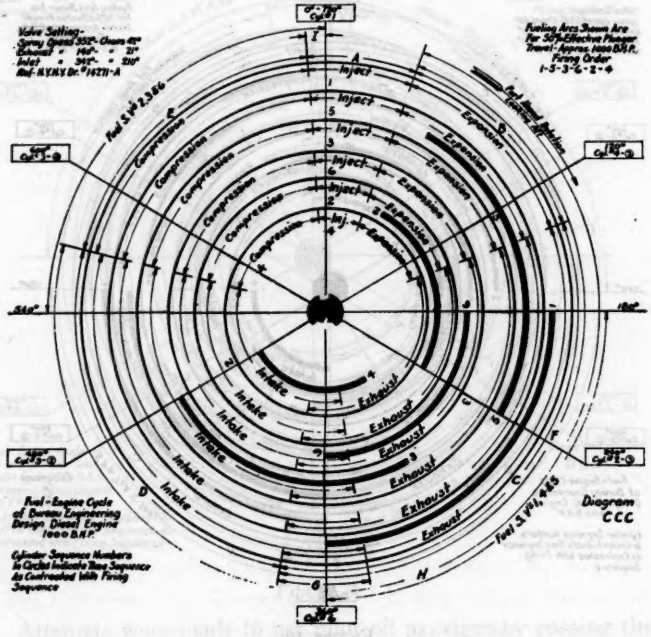


FIGURE 11.

(*b*) Exhaust valve failures are due principally to three causes:

(1) High temperature at point of exhaust opening in dangerously overloaded cylinders. Cylinders *normally* overloaded, having perfect combustion, should have high exhaust temperatures. But in this type of engine, it is possible, with

wide open throttles, to dangerously overload the cylinders. These engines are very commonly run with throttles wide open to make 380 R.P.M. This causes mean effective pressures to run extremely high in some cylinders, due to low R.P.M. and high fuel consumption per unit of piston displacement. Heavy black smoke is a most natural result. It should be here noted that more fuel can be burned at the low speed than at proper speed for the throttle setting, because, at the low speed, the spray valve is open for a longer time; thus, after the air has injected all the fuel in the spray valve, it continues to blow into the cylinder, supplying additional oxygen for combustion of the excess fuel.

(2) As set forth under the discussion of spray valves, a leaking spray valve will cause burning of exhaust valves, due to afterburning, if the spray valve has oil delivered to it at any point in the cycle between the closing of the spray valve and the closing of the intake valve. As was pointed out above, all cylinders of the R-21 to 27 class engine receive oil during this period.

(3) Both conditions set forth in paragraphs (1) and (2) above, cause smoke. The smoke carries carbon which deposits on the exhaust valve parts. This deposit in time causes valves to seat sluggishly and finally to stick open. This sluggish seating action causes excessive heating of the valve, due to ineffective seat cooling.

(*c*) Exhaust valve cages usually fail because of local overheating. This is caused by leaking valves. A valve usually develops one large leak. This large leak allows flame to blow through during working strokes. The flame impinges against the cage at one point. This one point is overheated and a crack results. Bad exhaust valve leaks are almost invariably accompanied by cage troubles. Cages also fail at the holding down lugs, when, due to overheat from excessive exhaust temperatures, or from leaking exhaust valves, the cage cooling becomes inadequate and undue expansion occurs. This causes the valve

to stretch up in relation to the better cooled cylinder head, thus breaking the lugs off. Of course, if the cage cooling system fails for any cause, the lugs will be broken off with normal exhaust temperatures.

(*d*) Cylinder heads fail around the exhaust cage seats because of local heating, as do the cages. In cases where the cage skirts are badly cracked, a close inspection will usually show that a small crack has started in the cage seat in the head. These cracks grow very rapidly.

(*e*) and (*f*) Pistons and cylinder heads are frequently punctured by an exhaust valve disc dropping off its stem and being caught by the piston as it approaches upper dead center. Several such occurrences were experienced by Submarine Division One at the Canal Zone during the past two years.

(*g*) The exhaust header failures, common to this type of engine, are, to a large extent, due to unbalanced loads in cylinders and afterburning, causing local heating strains. It was observed that engines in the worst condition had most header failures. The R-24 has had a cracked header for over a year. Special attention has been given to keeping the engine balanced by exercising unusual attention to the fuel measuring pumps. The crack has not increased in size since it was first noticed. Besides the uneven heating stresses from unbalanced power in various cylinders, the power unbalance causes undue header flexure due to undue weaving of the cylinders.

(*h*) Engine foundation trouble is largely dependent on power unbalance in various cylinders.

(*i*) Tie rod troubles are often believed to be foundation trouble, because the rods are invisible, and the cylinder flexure seems to come from the foundation. Overloaded cylinders overstress the rods, thus loosening them. As soon as they are loosened, friction and pounding commences. This rapidly increases the developed clearances.

(*j*) The vertical shafts of these engines give considerable trouble. This is due primarily to the lateral couples and pound-

ing set up by the fuel measuring pump driving eccentric. The steady bushing provided wears fast. The bushings are not renewed as often as they should be. For this reason heavy pounding obtains, and this results in the vibration loosening the keys in the gears and bolts in flanges.

(*k*) The governor failures will be discussed in detail under "Governor Effects" following. It is simply necessary here to recognize this as one of the troubles resulting from the present fuel distribution system.

The causes of the power unbalance, discussed above, are faults of design in the fuel measuring pumps. The major of these faults are:

(*a*) Double cyclic speed.
(*b*) Vacuum generation in the plunger chambers during the first half of the suction stroke.
(*c*) Group control of suction valves.
(*d*) Plungers are not floated.
(*e*) Difficulties of adjustments.
(*f*) Suction valves have excessive lift, causing spring trouble.
(*g*) Poor type of plunger packing.

Discussion of major faults listed above:

(*a*) The R-21 to 27 engine pumps run at the engine crank shaft speed. This, for the 4-cycle engine, is double the desired speed. The maximum safe R.P.M. for this 4-cycle engine is 420. This gives 420 pump cycles per minute. Pump slippage varies with the number of pump cycles per second. While reduction of the effective travel of the fuel pump plunger reduces slippage in the suction valves, it in no way affects the discharge valve slippage. With two discharges per cycle there are two slippage leaks per cycle. Thus, from the slippage standpoint, it is highly desirable to reduce the speed of the pump one-half. Further, the high speed allows less time to get the oil in motion to follow the plunger. There is little head of oil provided to overcome its inertia. The double cyclic

speed doubles the valve seatings per minute, thus doubles the wear. With halved speed, the plunger reversals will be slower, thus the valves will not pound their seats so hard because they will have longer time to seat. Halving the cyclic speed will reduce tappet wear and plunger and packing wear. Larger plunger diameters will make them a more practicable size.

(*b*) The fuel measuring pumps used on the R-21 to 27, V-1 to 3, and S-48 to 51 classes of Submarines are essentially alike in design. Figure 12 shows the control mechanism of the pump. The pump is of opposed type, there being groups of three plungers on each side carried by a yoke (*a*). The yokes are tied together by a tie-rod (*i*) on each side. The individual plungers, driven in groups, are controlled by suction valves whose control tappets are carried in similarly grouped rocker arms. These rocker arms are shown at (*g*). They are fulcrumed at (*j*) on an eccentric shaft which is controlled by the throttle. This, by raising or lowering the arms, simply changes the tappet clearance at (*c*). The lower ends of the rocker arms are held against the tappets by a spring (*b*), and as the guide rods have a constant travel, the rocker arms oscillate in a constant arc. The fulcrum height, regulated by the throttle, regulates the upper or individual tappet action. The rockers have an idle arc and a working arc. When the fulcrum position is lowest the least tappet clearance is provided, thus in this position the minimum idle arc is obtained. This necessarily means that the valve will be opened earlier in the stroke and seated later.

Referring to Figure 13, which is a diagrammatic arrangement of the standard reciprocation control, such as that in Figure 12, it is seen that the same lettering of parts is assigned as in Figure 12. This sketch shows a diagrammatic crank and connecting rod to drive the plunger. These represent the eccentric and rod which drive the pump in Figure 12. This type and Bureau type pumps use one-half of the discharge stroke for effective travel. The plunger travels in direction

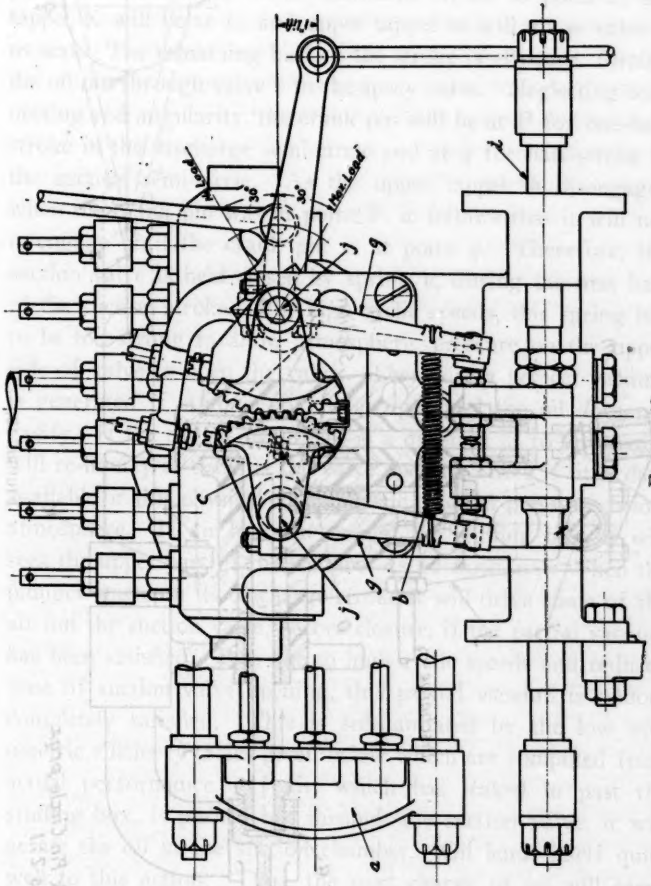


FIGURE 12.

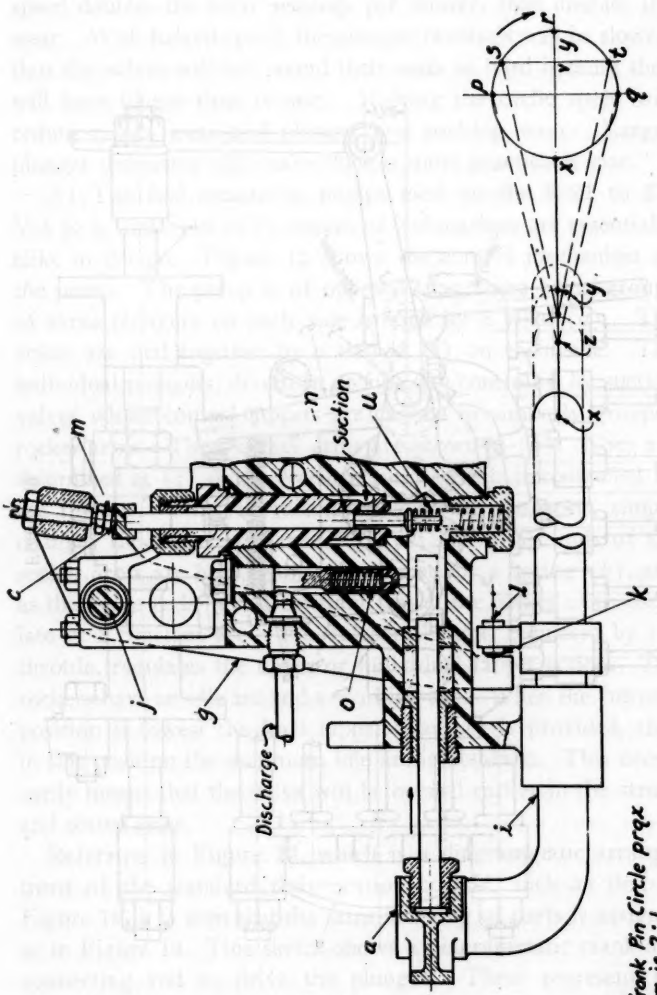

p
s
r
q
x
y
z
m
n
Suction
u
c
l
k
j
g
o
Discharge
i
a
Scale of Crank Pin Circle prqx
to Diagram is 2:1

FIGURE 13.

shown by arrow to discharge. Thus, (x) is the suction dead center of crank and (y) the delivery dead center. When plunger has traveled on its discharge stroke to point z, the tappet K will be at L, and upper tappet m will allow valve n to seat. The remaining half of the stroke is effective, forcing the oil out through valve o to the spray valve. Neglecting connecting rod angularity, the crank pin will be at P for one-half stroke in the discharge semi-circle and at q for half-stroke in the suction semi-circle. As the upper tappet m disengaged when the crank pin was at point P, it follows that it will not re-engage until the crank pin is at point q. Therefore, the suction valve is held closed by spring u, during the first half of the suction stroke. For high cyclic speeds, this spring has to be too strong to allow atmospheric pressure on the upper side of valve to open the valve. Therefore a perfect vacuum is generated if stuffing box be airtight and the oil does not gasify. If the oil gasifies, there is a question as to whether it will re-liquify as soon as the valve opens. There is little data available on this physical characteristic of oil at pressures below atmosphere. If air leaks in around the stuffing box, it will seek the upper part of the chamber as oil flows in. When the plunger starts on its discharge stroke it will drive some of the air out the suction valve, before closure, if the partial vacuum has been satisfied. But, due to high cyclic speeds and reduced time of suction valve opening, this partial vacuum is seldom completely satisfied. This is substantiated by the low volumetric efficiency tables given below which are computed from actual performance. If air, which has leaked in past the stuffing box, is pushed out through the suction valve, it will aerate the oil in the suction chamber. Oil lends itself quite well to this action. Thus, the next charge of oil will carry some of the discharged air back into the plunger chamber with it. When the suction valve closes, the pressure within the chamber builds up. The sharpness of this pressure rise should be, roughly, inversely proportional to the aeration degree. It

is evident, therefore, that air leaks affect the relative volumetric efficiencies of the different individuals' pumps, both directly and indirectly. If air leaks, oil leaks, valve lags and plunger cavitation are uniform, uniform volumetric efficiencies should obtain. But indicator card M.E.P.s indicate great variance. The plunger cavitation can be almost entirely stopped by low cyclic speeds, proper valve design and a valve control mechanism *which will positively open the suction valve at the beginning of the suction stroke at all times.* This will prevent any pressure reversals across the packing glands—pressure will always be outward.

Referring again to Figure 13, it will be seen that if the throttle is so set that the suction valve closes when the crank pin is at *s*, the effective travel will be ¼ and the valve will reopen on the suction stroke at *t*. Thus the suction valve will, in this case, be held open ¾ of the suction stroke. Similarly, if throttle be closed, the suction valve will seat only at point R to immediately reopen. Thus the suction valve will be held open throughout the suction stroke; but this obtains at no power condition only.

(*c*) As explained under (*b*) above, the pump is group controlled. Aside from the improper timing this provides, it necessitates group adjustments. To adjust the pump, it is usually jacked to its center of travel. The center is usually determined by a simple process of measuring the center of travel with a scale. The center thus found is prick punched; and another prick punch mark is placed on a stationary part. A tram is used in these two marks to get the pump centered. This method, though accurate enough for ordinary purposes, is not accurate enough for this purpose, as will be shown. Suppose a mistake of .010 inch is made in centering the pump. When the tappet clearance of the two rocker arms is set equal, one of them will be set .010 inch too close and the other .010 inch too much open. This means that one group of plungers will have .020 inch more effective travel than

the other group, even though the tappet clearance was accurately equalized. The firing order for one direction of rotation is 1–4–2–6–3–5 as shown in Figure 3. Pump plungers are grouped 1–2–3 and 4–5–6. Taking this grouping into the firing order, it is seen that alternate cylinders are affected. In bad cases, an exhaust intensity wave is noticeable which is quite regular. This effect is usually believed due to one cylinder misadjustment. Accordingly, an individual tappet adjustment is altered. This breaks up the rythm, and the tappet adjustment (either good or bad) is considered by the operator a proper cure for the uneven exhaust. The effective travel of the plunger of the fuel pumps of R-21 to 27 class is .625 inch. The .020 inch error pointed out above is equal to 3.2 per cent. This assumed that the throttle be wide open, but the error (absolute value) is constant at all throttle settings. Thus, if the throttle is set to deliver 1/10 of a full stroke of fuel, the effective travel should be .0625 inch. In this case the 0.20 inch error is equal to 31.8 per cent. If the volumetric efficiency of the pump is normally high, the throttle is not opened wide for full power. The 3.2 per cent given above is the minimum for .010 inch error in tramming the pump. In some cases the adjustment of these opposed pumps is made with throttles closed and pump on dead center. This requires the pump to be jacked to both dead centers to set the two groups. This method of adjustment is, particularly for reduced loads, the best adjustment to use. It has not the disadvantage of a cumulative tappet clearance error as has the method of setting clearances as set forth above. However, as rocker arm forces are not so direct and tappet faces are contacted at angles to each other, this method is not more reliable than the other, where considerable wear has taken place in the rocker arm eccentric shafts and on the tappets.

The indicator cards of the 2250 H.P. V-1 to 3 engines were examined for evidence of group unbalance. These cards are those taken during block tests of these engines. It therefore

must be assumed that the adjustments made to the fuel pump were at least as good as can be expected in service. Records are shown for engines Nos. 673 and 674. In the tables below, M.E.P.s for cylinders 1, 2 and 3 are averaged, as are those for 4, 5 and 6. On the line below the average is shown the "P.D." or percentage of difference in mean effective pressures. An arrow in each case points to the highest average. The letter in the center in each case shows the test on which cards were taken:

ENGINE NO. 673.

1	—	93		4	—	99	1	—	105	4	— 112
2	—	101.5	"A"	5	—	99.5	2	—	104	"B" 5	— 110
3	—	100		6	—	99	3	—	120	6	— 103.5
T		294.5				297.5	T.		329		325.5
Avg.		98.16 →				99.16	Avg.		109.6 ←		108.5
		P.D. = 1.							P.D. = 1.		
1	—	105		4	—	100	1	—	110	4	— 97
2	—	100	"A"	5	—	110	2	—	100.5	5	— 94
3	—	105		6	—	105	3	—	105	6	— 93.5
		310				315			315.5		284.5
Avg.		103.3 →				105	Avg.		105.2 ←		94.8
		P.D. = 1.7							P.D. = 10.6		

The group balance for this engine was good except on the last test shown. A new adjustment was evidently made for the last run, because groups are better balanced within themselves, but unbalanced with each other.

ENGINE NO. 674.

1	—	85		4	—	95	1	—	95	4	— 105
2	—	100	"A"	5	—	100	2	—	100	"A" 5	— 105
3	—	95		6	—	100	3	—	100	6	— 110
T		280				295			295		320
Avg.		93.3 →				98.3	Avg.		98.3 →		106.6
		P.D. = 5.2							P.D. = 8.4		
1	—	95		4	—	105	1	—	80	4	— 90
2	—	110	"B"	5	—	110	2	—	90	"E" 5	— 95
3	—	85		6	—	110	3	—	75	6	— 95
T		290				325	T		245		280
Avg.		96.6 →				108.3	Avg.		81.6 →		93.3
		P.D. = 12.2							P.D. = 14.2		

1	—	80			4	—	85	1	—	50	
2	—	85	"C"		5	—	85	2	—	70	"F"
3	—	65			6	—	85	3	—	50	
T		230					255	T		170	
Avg.		76.6					85	Avg.		56.6	
			P.D. = 10.9								P.D. = 17.6
1	—	65			4	—	75	1	—	45	
2	—	70	"D"		5	—	75	2	—	55	"G"
3	—	60			6	—	75	3	—	45	
T		195					225	T.		145	
Avg.		65.0					75	Avg.		48.3	
			P.D. = 15.4								P.D. = 24.2

4	—	65			4	—	60				
5	—	65			5	—	60				
6	—	70			6	—	60				
		200					180				
		66.6					60				

ANALYSIS TABLE TO NO. 674 COMPARISONS.

Test A.

Group 1	M.E.P.	98.3
Group 2	M.E.P.	106.6
Average		102.4
P.D.		8.4

Test B.

Group 1	M.E.P.	96.6
Group 2	M.E.P.	108.3
Average		102.4
P.D.		12.2

Test D.

Group 1	M.E.P.	65.0
Group 2	M.E.P.	75.
Average		70.0
P.D.		15.4

Test F.

Group 1	M.E.P.	56.6
Group 2	M.E.P.	66.6
Average		61.6
P.D.		17.6

Test A.

Group 1	M.E.P.	93.3
Group 2	M.E.P.	98.3
Average		95.8
P.D.		5.2

Test C.

Group 1	M.E.P.	76.6
Group 2	M.E.P.	85.
Average		80.8
P.D.		10.9

Test E.

Group 1	M.E.P.	81.6
Group 2	M.E.P.	93.3
Average		87.4
P.D.		14.2

Test G.

Group 1	M.E.P.	48.3
Group 2	M.E.P.	60.
		108.3
Average	M.E.P.	54.1
P.D.		24.2

SUMMARY.

Test.	*M.E.P.*	*P.D.*	*H.P.*	*R.P.M.*
A	95.8	5.2	2250	310
A	102.4	8.4	2250	310
B	102.4	12.2	2500	325
C	80.8	10.9	1289	310
D	70.0	15.4	1125	310
E	87.4	14.2	1910	271
F	61.6	17.6	1125	232
G	54.1	24.2	675	203

This engine shows greatest group unbalance of the five records available. As set forth above, under the discussion of group adjustment clearances, it is seen that the greatest differences (24.2 per cent) obtained at the lightest load. While the summarizing table does not indicate a true inverse progression of group unbalance to M.E.P., the trend is true to expectation.

(*d*) Referring to the view of the plunger yoke arrangement in Figure 12, it is seen that the plunger heads are fastened rigidly to the yoke. To be tight in the packing, these plungers should have not more than .002 inch clearance. The guide rods, however, must have at least .006 inch to run cool in the guides. These two needs with rigidly connected plungers are irreconcilable. As the pumps cannot be operated with too little clearance in the guide rods, the plungers and packing rapidly wear and cause leaks. The short connecting rod puts a considerable side thrust on the guide rods. This, through the rigid connection at the plunger heads, puts side thrust in plungers at the packing.

The R-21 to 27 and S-48 to 51 classes are the only type of fuel measuring pump in use in the Navy in submarines which has no provision for floating the plungers.

(*e*) Some difficulties of getting proper group adjustments were discussed above. Two tappet adjustments are required for each suction valve. That is, the group tappet and the individual tappets carried in the tappet yoke. It is difficult to get the individual tappets uniformly adjusted because of the tappet cap construction.

(*f*) As shown above under (*b*), the rocker arms have an idling arc and a working arc. The working arc plus the idling arc is a constant value. The working arc corresponds to the ineffective plunger travel. It therefore follows that as the ineffective travel increases, the lift of the valve increases, being maximum at maximum ineffective travel (or closed throttle position). Thus we have the maximum valve lift at no power and minimum at full power. Obviously, if proper

valve lift is provided for full power, it at reduced powers will be excessive. This causes undue inertia generation in the reciprocating valve, which requires heavier springs to prevent tossing of valve at low loads and high speeds.

(*g*) The plunger packing is poor. Primarily, it is too short, being only 2 inches, whereas the packing cylinders used in the Bureau type pump is 4¾ inches. The lapped type of packing allows no personal element of assembly or adjustment. It is the ideal form of packing, but must have floated plungers.

All of the above faults contribute to low and erratic volumetric efficiencies. Low volumetric efficiencies should not affect the functioning of an engine if the efficiency were absolutely uniform in all cylinders. But this is not the case, and in view of the above discussed faults it cannot reasonably be expected that it should be. Leaks or valve lags cannot be assumed uniform. The suction valves have packed stems. These, like the plungers, are subject to the personal element of assembly and adjustment. Uneven packing adjustments cause varying suction valve closure lags. Packing these stems is a cardinal mistake. They should be lapped fits, or pumps should be fitted with float chambers, preferably the latter.

The records of the V-1 to 3 engines, and those of the S-48 to 51 engines have been searched for effects of volumetric efficiencies. Some of the results are tabulated below. The efficiencies were computed from the formulae following:

Displacement of pump per hour in gallons =

$$\frac{D^2 \times .7854 \times Te \times Cp \times 60 \times 6 \times F \times F_2}{231}$$

where:

D=plunger diameter
Te = plunger travel (effective)
Cp = pump cycles per minute
6 = number of plungers
60 = minutes
F = Factor for liquid compression

F_2 = Factor for liquid expansion and contraction due to heat
231 = cubic inches per gallon

The factor F was considered equal to 1 because the liquid compression (with no aeration) for the discharge pressure of the pump is approximately .00056. The factor F_2 was considered equal to 1 because, while no data was taken on temperature of the oil at the pump, the Senior Member of the Trial Board believed 60 degrees F. to be a good average. This is the standard temperature of tables used.

$$\text{Volumetric Efficiency} = \frac{\text{Pump displacement}}{\text{Fuel in gallons actually consumed}}$$

U.S.S. V-1 Test "A" Starboard 5-6 November, 1923
Average throttle position = 6.76
Corresponding displacement = Y gallons
Fuel oil gravity—Be 29 specific .8805
Pounds per gallon 7.332
Fuel consumption per engine hour X pounds.

$\frac{X}{7.332}$ = Z gallons, $\frac{Z}{Y}$ =86.1 per cent Volumetric efficiency.

Exhaust—dark gray 25 hours.
Exhaust — dark 1 hour H.P. 2250; R.P.M. 310

The following summarizing table shows the result for all records studied, using the above method:

TESTS "A" (FULL LOAD) 2250 H.P.

Boat	*Engine*	*Pump Efficiency per cent*	*Exhaust*
V-3	Starboard	105.8	Gray
V-1	Starboard	86.1	Dark Gray
V-2	Starboard	85.8	Gray
V-3	Port	75.62	Black
V-1	Port	75.27	Black
V-2	Port	73.35	Black

TESTS "B" (OVERLOAD) 2500 H.P.

V-3	Starboard	107	Dark
V-2	Starboard	93.3	Black
V-1	Port	82.3	Black
V-2	Port	81.6	Black
V-1	Starboard	80.6	Black
V-3	Port	75.62	Black

Tests C, D, E, F and G, being light loads do not show any smoke; therefore, no comparisons can be drawn regarding the pump volumetric efficiencies.

Analysis table of tests "A" (2nd paragraph above) shows the average volumetric efficiencies of the fuel pumps of the six "V" boat engines, listed in sequence according to efficiency values. The first one showing an efficiency exceeding 100 per cent is obviously wrong. This is probably due to the throttle linkage being improperly connected. It is important to note that where the efficiencies are above 80, the exhaust is gray, but where efficiencies fall below 80, the exhaust is black. It should further be noted that the highest efficiency for tests "B" gives a "dark" exhaust as compared with the others which are black.

As stated above, it cannot be assumed that efficiencies of all six individual pumps in one engine will be uniform where the efficiency for the engine is low. Applying the law of average, it is evident that some of the six individual pumps will retain an initial efficiency, whereas others will drop. This will cause the throttle to be opened more for development of the same horsepower. Obviously this will overload some cylinders, causing smoke. This means afterburning with attendant burned exhaust valves and cages in 4-cycle engines, and cracked cylinders, heads and pistons in 2-cycles. It is marvelous that a wrist pin can be lubricated at all where the piston is so hot that the metal in it fails. The heat decomposes the oil and thus the film is broken. No kind of bearing metal can withstand friction unless a stable oil film is provided to separate the surfaces.

S-48 TO 51 CLASS.

The volumetric efficiencies of the test runs of the S-48 to 51 class of engines were not computed. The indicator cards for the S-48 to 51 class showed extreme differences between cylinders in all engines. Differences are so great that group unbalances can not consistently be determined. Refer to Figures M and N. Each of these figures show accurate tracings of the indicator cards, the recorded M.E.P.s, H.P.s, engine and cylinder numbers. Added to each figure is a table (reproduced below) having four columns; No. 1 being the cylinder, No. 2 the developed horsepower of the cylinder, No. 3 the percentage of the engine total load carried by the cylinder, No. 4 the percentage of average cylinder load carried by the cylinder.

ENGINE NO. 581—FIGURE M.

1	287.5	19.15		114.9
2	237.5	15.85		95.1
3	175.0	11.67	R.P.M.	70.0
4	212.5	14.15	349	84.8
5	350.	23.34		140.0
6	237.5	15.84		95.2
Total	1500.0	100.00		

This power balance is bad. Cylinder No. 5 carried 40 per cent greater load than a balanced load. As this test was an overload run, the No. 5 cylinder was loaded 40 per cent higher than the maximum for which designed.

ENGINE NO. 581—FIGURE N.

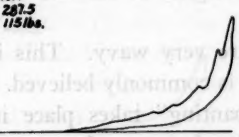
1	70	18.87		113.2
2	63	16.98		102.0
3	35	9.43	R.P.M.	56.5
4	49	13.22	195	79.2
5	91	24.52		147.1
6	63	16.98		102.0
Total	371	100.00		

Engine No. 581 - I.H.P. 1500
R.P.M. 349 - Overload

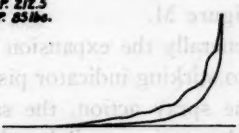
Fig. M

Refer To Discussion

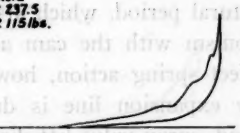
Cyl. No. 1
I.H.P. 287.5
M.I.P. 115 lbs.



Cyl. No. 4
I.H.P. 212.5
M.I.P. 85 lbs.

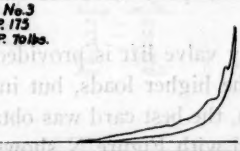


Cyl. No. 2
I.H.P. 237.5
M.I.P. 115 lbs.



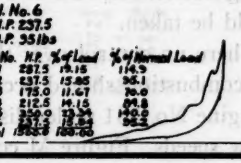
Cyl. No. 5
I.H.P. 350.0
M.I.P. 140 lbs.

Cyl. No. 3
I.H.P. 175
M.I.P. 70 lbs.



Cyl. No. 6
I.H.P. 237.5
M.I.P. 95 lbs

Cyl. No.	H.P.	% of Load	% of Normal Load
1	287.5	19.15	114.9
2	237.5	15.85	95.1
3	175.0	11.67	70.0
4	212.5	14.15	84.8
5	350.0	23.34	140.0
6	237.5	15.84	95.0
Total	1500.0	100.00	

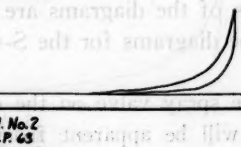


Engine No. 581 - I.H.P. 371
R.P.M. 195

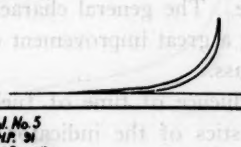
Fig. N

Refer To Discussion

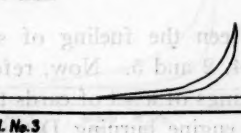
Cyl No. 1
I.H.P. 76
M.I.P. 50 lbs.



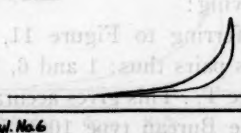
Cyl. No. 4
I.H.P. 49
M.I.P. 35 lbs



Cyl. No. 2
I.H.P. 63
M.I.P. 45 lbs.



Cyl. No. 5
I.H.P. 91
M.I.P. 65 lbs.



Cyl. No. 3
I.H.P. 35
M.I.P. 25 lbs

Cyl. No. 6
I.H.P. 63
M.I.P. 45 lbs.

Cyl. No.	H.P.	% of Load	% of Normal Load
1	76	19.97	113.2
2	63	16.54	197.0
3	35	9.45	56.6
4	49	13.12	78.8
5	91	24.02	137.1
6	63	16.90	100.0
Total	377	100.00	

The power balance for run represented by Table N is bad, too; but, due to lighter engine loading, should not cause such heating and over-stressing in the engine as for the overload in Figure M.

Generally the expansion lines are very wavy. This is not due to sticking indicator piston, as is commonly believed. Due to the spray action, the same "panting" takes place in the cylinder as in an oil burning boiler furnace. In some cases the spray valve springs have a natural period, which at some speeds falls into multiple synchronism with the cam action. This effect may obtain with perfect spring action, however. To determine whether a wavy expansion line is due to spring vibration in its natural period, spray valve lift diagram should be taken.

Where no control over the spray valve lift is provided, the best combustion should occur at the higher loads, but in case of engine No. 581 under discussion, the best card was obtained at low speeds. Figure M compared with Figure N shows this to be true, the former figure showing a high power run.

Figures R and S show two sets of the cards of the V-1 to 3 engines which were discussed under subject of group unbalance above. The general characteristics of the diagrams are fair, being a great improvement over the diagrams for the S-48 to 51 class.

Influence of time of fueling the spray valve on the characteristics of the indicator cards will be apparent from the following:

Referring to Figure 11, it is seen the fueling of spray valves pairs thus: 1 and 6, 3 and 4, 2 and 5. Now, refer to Figure T. This gives accurate tracings of a set of cards taken on the Bureau type 1000 B.H.P. engine burning Diesel oil. These cards were taken in the year 1922 at the Navy Yard, New York, while tests were being run burning Bunker "C" oil. Number two cylinder has an extremely high maximum pressure peak. This was found due to the working piston

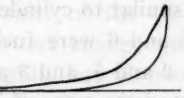
Engine No. 674 - B.H.P. 673
R.P.M 202 - Load Test G

Fig. R.

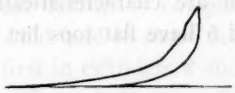
Refer To Discussion Page*

Cyl. No. 1
I.H.P. 182.3
M.I.P. 45 lbs.

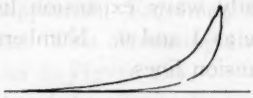
Cyl. No. 4
I.H.P. 243.0
M.I.P. 60 lbs.



Cyl. No. 2
I.H.P 222.8
M.I.P. 55 lbs.

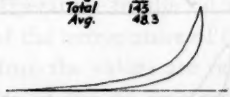


Cyl. No. 5
I.H.P. 243.0
M.I.P. 60 lbs.



Cyl. No. 3
I.H.P. 182.3
M.I.P. 45 lbs.

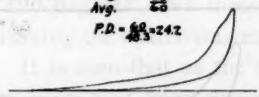
Group	M.I.Ps
#1	45
#2	55
#3	45
Total	145
Avg.	48.3



Cyl. No. 6
I.H.P 243.0
M.I.P 60 lbs.

Group	M.I.Ps.
#4	60
#5	60
#6	60
Avg.	60

P.D. = $\frac{60}{48.3}$ = 24.2



Engine No. 674 - H.P. 2500
R.P.M. 325

Fig. S

Refer To Discussion Page

Cyl. No. 1
I.H.P. 619
M.I.P. 95 lbs.

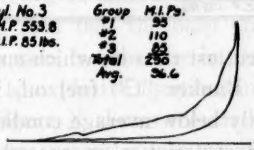
Cyl. No. 4
I.H.P. 684
M.I.P 105 lbs.

Cyl. No. 2
I.H.P. 716.8
M.I.P. 110 lbs.

Cyl. No. 5
I.H.P. 716.8
M.I.P. 110 lbs.

Cyl. No. 3
I.H.P. 553.8
M.I.P. 85 lbs.

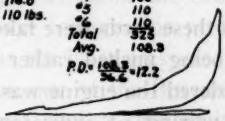
Group	M.I.Ps.
#1	95
#2	110
#3	85
Total	290
Avg.	96.6



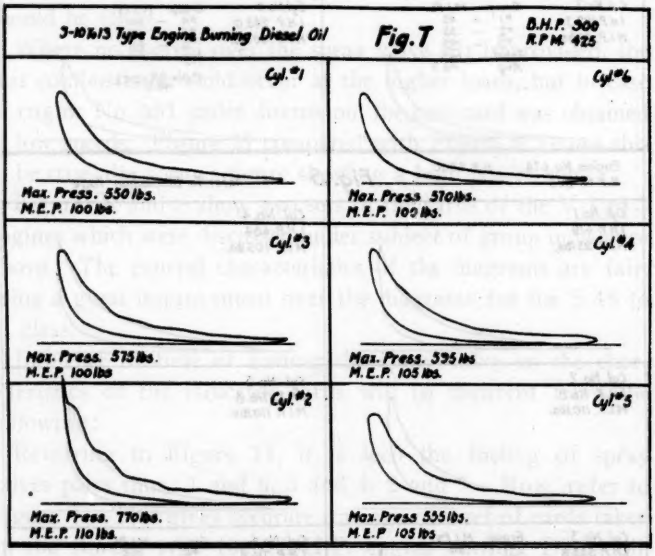
Cyl. No. 6
I.H.P. 716.8
M.I.P. 110 lbs.

Group	M.I.Ps.
#4	105
#5	110
#6	110
Total	325
Avg.	108.3

P.D. = $\frac{108.3}{96.6}$ = 12.2



being cracked about three-fourths of the way around in the upper ring groove. This large leak passed large quantities of lubricating oil into the cylinder, which, on compression, detonated. Were it not for this peak, this card should have a top similar to cylinder No. 5. It was seen above that cylinders 1 and 6 were fueled in the same cyclic time. Similarly so are 2 and 5, and 3 and 4. In Figure T it is seen the card characteristics pair off in the same manner. Numbers 1 and 6 are the best cards and almost identical. Numbers 3 and 4 have slightly wavy expansion lines, but are characteristically the same as 1 and 6. Numbers 3 and 5 have flat tops but wavy expansion lines.



As these cards were taken between test runs on which engine was being pushed rather hard on Bunker "C" fuel oil, it is considered the engine was in slightly below average condition. The pairing of indicator card characteristics in accordance

with fuel diagram characteristics, as set forth above, does not hold true in all cases. In fact, it is obvious that cards have to be generally very good to make such small differences apparent.

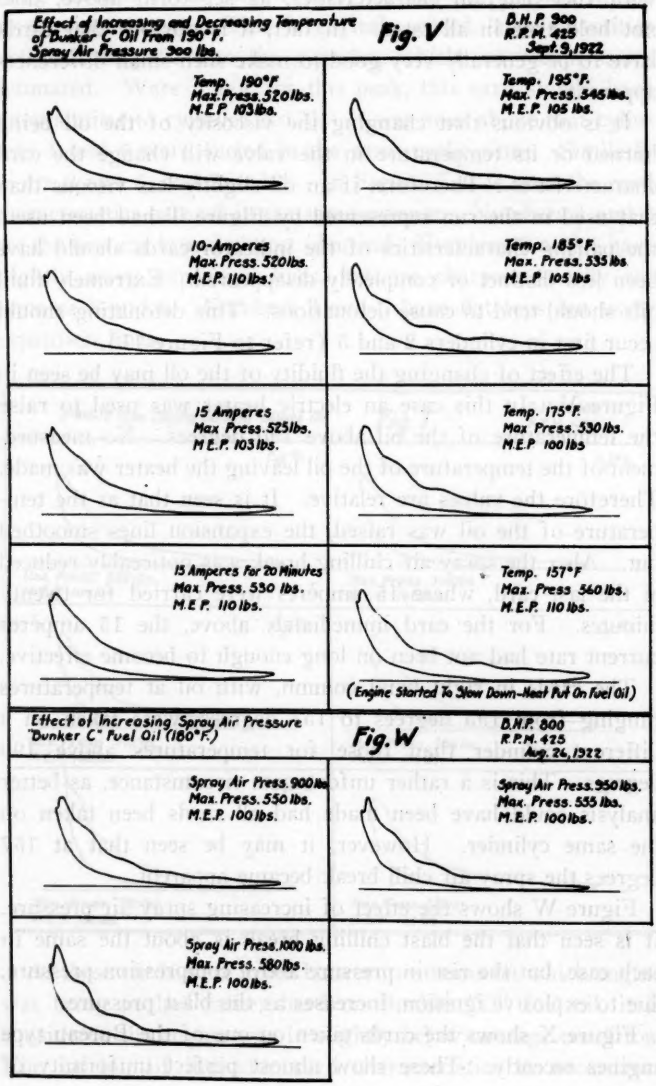
It is obvious that changing the viscosity of the oil being burned or its temperature in the valve will change the card characteristics. Therefore, if an oil slightly less viscous than that used in the run represented by Figure T had been used, the pairing characteristics of the indicator cards should have been less distinct or completely disappeared. Extremely fluid oils should tend to cause detonations. This detonating should occur first in cylinders 2 and 5 (refer to Figure 11).

The effect of changing the fluidity of the oil may be seen in Figure V. In this case an electric heater was used to raise the temperature of the oil above 190 degrees. No measurement of the temperature of the oil leaving the heater was made. Therefore the values are relative. It is seen that as the temperature of the oil was raised, the expansion lines smoothed out. Also, the spray air chilling break was noticeably reduced in the last card, where 15 ampères were carried for twenty minutes. For the card immediately above, the 15 ampères current rate had not been on long enough to become effective.

The cards in right hand column, with oil at temperatures ranging from 195 degrees to 157 degrees, were taken on a different cylinder than those for temperatures above 190 degrees. This is a rather unfortunate circumstance, as better analysis could have been made had all cards been taken on the same cylinder. However, it may be seen that at 157 degrees the spray air chill break became apparent.

Figure W shows the effect of increasing spray air pressure. It is seen that the blast chilling break is about the same in each case, but the rise in pressure above compression pressure, due to explosive ignition, increases as the blast pressure.

Figure X shows the cards taken on one of the Bureau type engines recently. These show almost perfect uniformity of


Effect of Increasing and Decreasing Temperature of Dunker C" Oil From 190°F.
Spray Air Pressure 900 lbs.
Fig. V
B.H.P. 900
R.P.M. 425
Sept. 9, 1922
Temp. 190°F
Max. Press. 520 lbs.
M.E.P. 103 lbs.
Temp. 195°F.
Max. Press. 545 lbs.
M.E.P. 105 lbs.
10-Amperes
Max. Press. 520 lbs.
M.E.P. 110 lbs.
Temp. 185°F.
Max. Press. 535 lbs
M.E.P 105 lbs
15 Amperes
Max Press. 525 lbs.
M.E.P. 103 lbs.
Temp. 175°F.
Max Press. 530 lbs.
M.E.P 100 lbs
15 Amperes For 20 Minutes
Max. Press. 530 lbs.
M.E.P. 110 lbs.
Temp. 157°F
Max. Press 560 lbs.
M.E.P. 110 lbs.
(Engine Started To Slow Down-Heat Put On Fuel Oil)
Effect of Increasing Spray Air Pressure "Dunker C" Fuel Oil (180°F.)
Fig. W
B.H.P. 800
R.P.M. 425
Aug. 24, 1922
Spray Air Press. 900 lbs
Max. Press. 550 lbs.
M.E.P. 100 lbs.
Spray Air Press. 950 lbs.
Max. Press. 535 lbs.
M.E.P. 100 lbs.
Spray Air Press. 1000 lbs.
Max. Press. 580 lbs.
M.E.P. 100 lbs.

characteristics, showing the relation of fuel viscosity to spray valve resistance was perfect. However, slight changes in spray roller clearance on some cylinders is sometimes necessary for these results. This clearance is made with the engine warmed up and running. The clearance is then shifted to correct the cards as necessary.

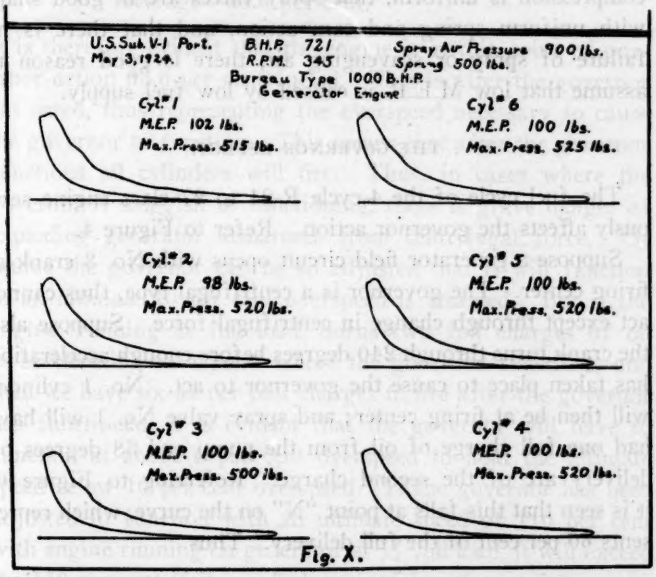


Fig. X.

The original design of the Bureau engine used different oil lead connection arrangement to the fuel measuring pumps than the connections now used. The two groups were connected thus (right to left): 1, 2, 3–4, 5, 6. But in all these engines the leads have been changed, so that they are now connected 1, 4, 5–2, 3, 6. At the time the design was made up it was assumed the cyclic time of fueling the valves made no difference, but later this assumption was found untrue.

The effect of the 1, 2, 3–4, 5, 6 grouping may be found by use of Figure 10.

In this discussion no attempt has been made to discuss indicator cards, except wherein they are incidentally affected by the fuel system cycle. There are many factors affecting indicator cards in a Diesel engine, and a low M.E.P. in a cylinder does not necessarily indicate that that cylinder is not getting its quota of fuel. However, if we know the valve timing and compression is uniform, that spray valves are in good shape with uniform spring and cam action, and that there is no failure of spray or scavenging air, there is good reason to assume that low M.E.P. is caused by low fuel supply.

THE GOVERNOR EFFECT.

The fuel cycle of the 4-cycle R-21 to 27 class engine seriously affects the governor action. Refer to Figure 4.

Suppose a generator field circuit opens with No. 3 crank at firing center. The governor is a centrifugal type, thus cannot act except through change in centrifugal force. Suppose also the crank turns through 240 degrees before enough acceleration has taken place to cause the governor to act. No. 1 cylinder will then be at firing center; and spray valve No. 1 will have had one full charge of oil from the pump and 38 degrees of delivery arc of the second charge. Referring to Figure 6, it is seen that this falls at point "N" on the curve, which represents 60 per cent of the full delivery. Thus

$$\frac{100 \text{ plus } 60}{200} =$$

80 per cent of a total charge has been delivered. Nos. 4, 2, 6 and 3 cylinders will have all of one stroke's delivery, or 50 per cent of a total charge. Cylinder No. 5 will have fired, and as the fueling began about 10 degrees before end of injection, will have 80 degrees of delivery angle, which on Figure 6 shows 83 per cent of the stroke, or 41.5 per cent of a total charge. Therefore spray valves are fueled as follows:

No. 1—80 per cent of full charge.
No. 2—50 per cent of full charge.
No. 3—50 per cent of full charge.
No. 4—50 per cent of full charge.
No. 5—41.5 per cent of full charge.
No. 6—50 per cent of full charge.
Average—53.6 per cent.

It is therefore evident that the engine is at this point of governor action 53.6 per cent fueled. This is after the governor has acted, thus representing the overspeed necessary to cause the governor to function. This means that after the governor functions all cylinders will fire. Thus, in cases where the governor is sluggish in functioning, there is grave danger of exploding generator armatures from centrifugal force. Of course the governor can be so adjusted that it will function so an *ultimate* 10 per cent overspeed is attained. But if the engine, running at full load, burns two full charges of oil in accelerating to a point where the governor functions, and then we have six 53 per cent charges to fire after the governor has functioned, it is evident that the governor will have to function at about 4 per cent overspeed to hold the ultimate speed below 10 per cent overspeed. If the governor has been adjusted to function with an ultimate speed of 110 per cent with engine running on generator at ½ full load, it will exceed the 110 per cent if it is called upon to function at full power. This is because that while the governor is adjusted to function for the same *number* of following cycles of oil for all speeds, the amount of oil delivered per cycle depends on the load at the time the generator functions. Thus, with the governor set to function at ½ load (indicated), and with the engine working at full load, the spray valves which supply cylinders, after the governor action, have received twice as much fuel as that for which the governor was adjusted. With the governor adjusted at ½ load to function at 110 per cent engine speed, more power impulses will be required, after the loss

of load, to increase the speed to the 10 per cent tripping point, than in case of full load; but this does not effect the maximum speed. The percentage ratios used are simply to show effects—somewhat higher ones actually obtain in service. Governors should thus be adjusted at full load, but they then function very close to the maximum operating speed at reduced loads. Failures of these governors are not unknown. The writer saw an armature removed from the R-25 which had exploded because the governor action was too sluggish. Several armature coils were thrown out and wedged between the armature and pole pieces. This, in addition to causing work, delay and expensive repair, sets up very heavy stresses in the engine, due to abrupt stopping.

A fuel measuring pump designed to fuel only one spray valve at a time will correct the governor faults discussed above to a material degree; and will make overspeed governors much more reliable.

Centrifugal governors, however, have marked tendencies to hunt. Combination of inertia weights with centrifugal weights are necessary for perfect governor action, especially at high rotative speeds.

SUMMARY.

The natural result of failure of a fuel measuring pump to function properly, unless it be adjustment unbalance, is lowered volumetric efficiency. If the volumetric efficiency is lowered, it is undeniable evidence that there is something wrong in at least one of the individual pumps. Loss in volumetric efficiency is certain when a materially greater than normal throttle opening is required to carry a certain load burning a normal amount of fuel per brake horsepower hour. The volumetric efficiency of a good pump should consistently stay above 90 per cent; and in the Bureau type pump it does.

The Bureau type pump maintains high volumetric efficiency because of five major factors:

(*a*) Plunger packing is ideal.

(*b*) Vacuum generation in plunger chambers is reduced. It is not totally eliminated, however, as about 24 inch vacuum is generated to open the suction valves automatically.

(*c*) Valve design is superior in type and location.

(*d*) The cyclic speed of the pump is the minimum.

(*e*) The suction valves are not hampered in action by stem packing.

Group unbalance is a most serious factor in opposed types of pumps.

Group suction valve control forces non-uniformity in fueling cycles which has adverse affects except where ideal conditions obtain.

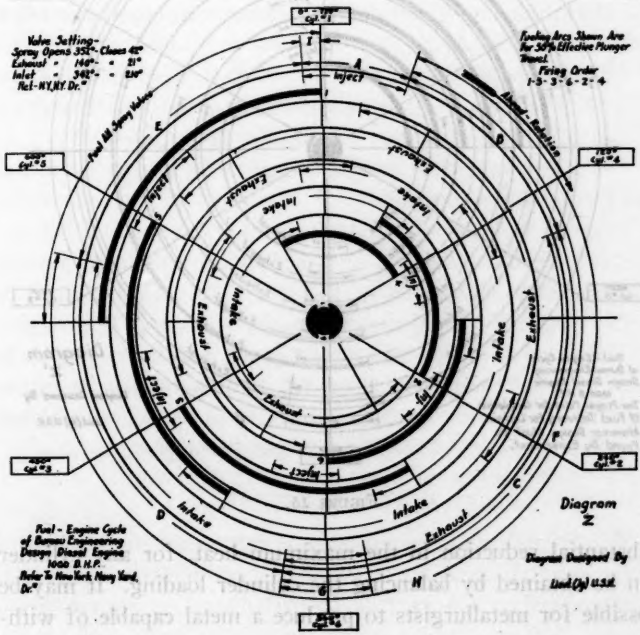


FIGURE 14.

It seems futile to try to lubricate wrist pins in overheated pistons. To attempt to especially cool the wrist pin with piston cooling oil should result in more cracked pistons. Further, it is doubtful if the wrist pin lubrication should be improved, because shifting the supply of cooling oil to the pin should cause a hotter piston head and greater heat flow, by conduction and radiation, to the wrist pin. If too much heat is *generated within the cylinder,* it must be absorbed. But it appears a

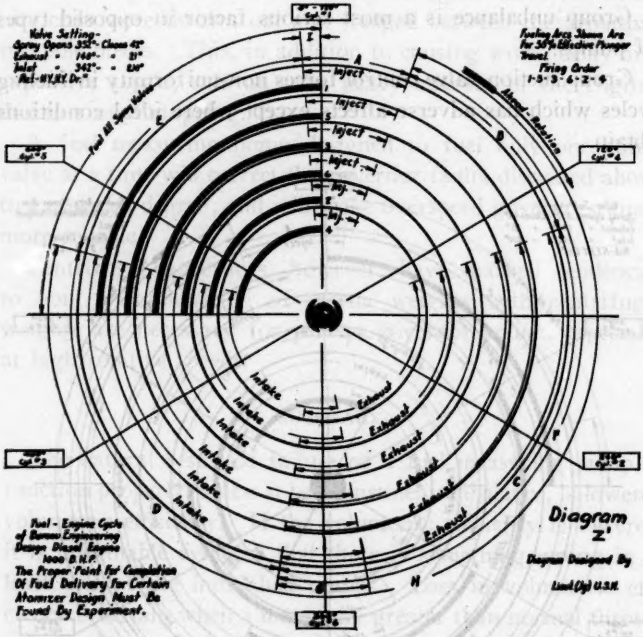


FIGURE 15.

substantial reduction in the maximum heat, for any cylinder can be obtained by balancing the cylinder loading. It may be possible for metallurgists to produce a metal capable with-

standing the heats now developed; but this metal will not solve the problem of oil film breakdown due to decomposition from excessive heat.

Figures 14 and 15 show the fuel-engine cycle of a pump designed to meet the timing requirements set forth above. The writer recently witnessed the performance of the engines in a Diesel-electric tug of the Pennsylvania Railroad. These engines are equipped with timed fuel pumps as above and the governor action was excellent.

BALL BEARINGS FOR MOTORS.*

By C. Huey, Material Engineer,
Navy Yard, New York.

The question as to whether ball bearings should or should not be used for motor bearings has been raised in connection with the purchase and operation of motors on Naval vessels. The present status of ball bearings, as applied to the Naval Service, is that it is practically optional as to whether they are specified or as to whether the manufacturer or contractor offers or supplies ball bearings.

The proper solution or attack of the question, therefore, resolves itself into a discussion or proof as to the following:

(*a*) Are sleeve bearings suitable for Navy motors?

(*b*) Are sleeve bearings, as supplied, of the most suitable design and manufacture?

(*c*) Are ball bearings suitable for Navy motors?

(*d*) Are ball bearings, as supplied, of the most suitable design and manufacture?

(*e*) Are ball bearings more suitable than sleeve bearings?

The scope of this paper will not appear as a direct answer to these hypothetical questions, but the advantages, disadvantages, limitations and some design data will be presented with a view to assisting in the drawing of a proper conclusion.

BEARINGS, GENERAL.

Manufacturers of Navy motors at present writing are limited in the design of bearing to the following quoted from general specifications on motors: "Ball or roller bearings are preferred.

*Based on investigation for and the report made to Bureau of Engineering, Navy Department.

Where ball or roller bearings are used, the design shall be such as to facilitate removal. Grease lubrication will be used unless otherwise specifically approved. Compression grease cups or other approved means for refilling shall be provided. Bearings, if of the sleeve type, shall be provided with approved babbitted or bronze linings, lubricated by balanced soft metal oil rings which shall run true. The following are approved compositions:

	Bronze	*Babbitt*
Copper, per cent	82 —84	3— 4.5
Tin, per cent	12.5—14.5	88—89.5
Zinc, per cent	2.5— 4.5	..—....
Antimony, per cent	...—....	7— 8

Bearings shall function satisfactorily when axis of shaft is inclined continuously at an angle of 10 degrees. The ultimate observable temperature permitted is 80 degrees C., while the maximum temperature rise permitted at 50 degrees C. room temperature, is 30 degrees C." The foregoing extract from the specifications embodies all the information that the designer of the motor has to work on. It is apparent that, excluding the limitation of allowable temperature and chemical analysis of the composition of sleeve bearings, the designer has unrestricted latitude in regard to the diameter, length, and looseness of fit of sleeve bearing. The tightness of fit is restricted by the allowable temperature rise. With ball or roller bearings, temperature rises are rarely, if ever, a disturbing factor for the designer. Therefore, it may be safely said that the manufacturer has no restrictions whatsoever in the design or quality of ball or roller bearings for Navy motors.

It is believed that sleeve bearings are fairly well established in Naval Service. Regardless of the assumption that little may be known as to the advantages or disadvantages, it is a fact that sleeve bearings during the past have carried the load of Navy motors under all conditions of service and are generally understandable by the personnel, as to maintenance, adjustments and replacements. In facing the ball or roller bearing

situation, conservative engineers view the ball bearing motor in a somewhat similar manner as they did turbine engines when these first entered the field against reciprocating engines. When viewing the operation of the pioneer turbine, the unspoken attitude was, "when will the thing fall apart?" Due to the fundamental superiority of the turbine, for electrical generating sets, the reciprocating engines for this purpose became obsolete as the better understanding and perfection of the turbines was accomplished. It is believed the present status of ball bearings versus sleeve bearings for specific uses is greatly analogous.

Although ball bearings have been in commercial use for many years, and are generally successful in many fields of application, the important details of design and the limitations of uses have been left to the manufacturer. The user, in most cases, and no less so in the case of the Navy, has depended upon the manufacturer and trusted that satisfactory results would be obtained. Under such a condition, uniformly reliable bearings could not be expected. By properly detailed specifications, this uncertainty could be greatly reduced and in time eliminated. It is essential that the user guard against proven faulty design and application.

SLEEVE BEARINGS.

With the foregoing paragraphs definitely pointing the way to ball bearings, it is desirable to tabulate the advantages and disadvantages of sleeve bearings so that the reasons for retaining sleeve bearings will be found among the "advantages," while the reasons for specifying or superseding the sleeve bearings by ball bearings will be found among the "disadvantages."

ADVANTAGES OF SLEEVE BEARINGS.

(*a*) Replacements can readily be manufactured on shipboard.

(*b*) Definite specifications as to materials can readily be checked.

(*c*) Running clearance between the shaft and the bearing can readily be controlled and determined by temperature rises and thickness gauges.

(*d*) Deterioration of the bearing from use or abuse, can readily be determined and estimated. The probable additional useful life can be estimated.

(*e*) Bearings may continue to be used after excessive wear, providing the armature does not rub on the pole pieces.

(*f*) Salt water or moisture in the bearing or in the oil will not seriously affect the operation, as the bearing is of non-corrodible material, and the corrosion of the steel shaft would have to be of large amount to be serious.

DISADVANTAGES OF SLEEVE BEARINGS.

(*a*) Increases the over-all length of the machine.

(*b*) End thrust cannot be properly taken or lubricated. Increase in friction load and additional bearing heat results from end thrust.

(*c*) Oil grooves in bearing are difficult to accomplish neatly and effectively, especially on a one-piece bronze of small diameter and long length.

(*d*) Practically impossible, except on oil pressure lubrication, to maintain the oil film on the entire area of the bearing, especially at the ends. This condition is aggravated if the motor is tilted from the horizontal.

(*e*) Ring oilers, which are used on practically 100 per cent of Navy motors, are difficult to manufacture so as to be well balanced dynamically and so as to efficiently circulate the oil.

(*f*) Ring oilers are readily damaged, and when damaged may cease to revolve and result in a dry and burnt bearing.

(*g*) Ring oilers if of too hard material will wear the shaft; if of too soft material will wear themselves. The resultant metal particles pass through the bearing surfaces.

(*h*) If the oil is allowed to fall below a certain level the oil rings will dance and throw the remaining oil out of the bearing. This oil throwing also results from a rough shaft.

(*i*) If oil, which has not a low "cold" test, is used in extreme cold weather, the bearing may not receive lubrication until damage is done to the bearing surfaces.

(*j*) On account of the fact that sleeve bearing will continue to function and support the load after excessive wear, the armature may be damaged by contact with the field pole pieces. Also commutation is affected by the armature being out of magnetic center.

(*k*) To reduce end thrust wear, considerable end thrust clearances are necessary. This allows the armature to rock end wise, in rolling or pitching of ship.

(*l*) A sleeve bearing will seize or burn out if emery or other serious foreign matter is accidentally or surreptitiously placed in the bearing.

(*m*) As all motors run at a fairly high shaft speed, continuous oil lubrication is necessary. This involves the use of oil wells, overflow holes, and oil sight level glasses, so that a predetermined oil level may at all times be maintained in the bearing for the purpose of feeding the oil rings. In addition to considerable mechanical damage to the oil level glasses and possible leakage of oil from the pipe fittings used on the oiling system, the bearings usually are supplied with additional oil from time to time by the operators. Due to the fact that the motor is usually in a dark corner and the oiling places somewhat inaccessible to reach with an oil can, considerable flooding and slopping of the oil is a usual result.

(*n*) On the driving end of an enclosed motor the suction effect of the revolving armature exerts a strong pull on the oil supply in the bearing and it is difficult to prevent the oil from spreading over the armature and field coil windings. On the commutator end, the surplus oil or slopped oil travels along to the commutator and brush rigging. This, together with carbon and copper dust, results in sticky brushes and carbonized insulation between bars of the commutator and at the commutator end clamp ring.

(*o*) Due to the high fluid friction of oil, there is a practical limit of speed at which a sleeve bearing may be operated.

(*p*) The mechanical efficiency of sleeve bearings is very low compared with ball bearings. On tight fitting sleeve bearings, or on split type sleeve bearings in which the bearing cap is not properly shimmed, the losses in the bearing may readily be high enough to overload the motor.

Assuming that bearings for motors are undergoing a transitory stage, and that sleeve bearings will be in evidence for some time to come, it is desirable as an improvement on the situation, to enlarge on the present specifications as to the permissible design dimensions, and to definitely limit the manufacturers as to whether bronze or babbitt lined bearings are to be furnished.

VALUE OF ANTI-FRICTION METALS FOR SLEEVE BEARINGS.

In order to bring to light the reasons for the optional use of bronze or babbitt lined sleeve bearings, and to dispel the prevailing idea that babbitt lined bearings have less friction, and therefore less heating than bronze bearings, the investigations of writers on the subject were consulted. The conclusions drawn from this information which check with practical experience are stated as follows:

The various white metals available for lining brasses do not afford coefficients of friction lower than can be obtained with bare bronze, but they are less liable to overheating because of the superiority of such material over bronze, to withstand abrasion or crushing without excessive increase in friction. Bronze, babbitt and other soft white alloys, have substantially the same coefficient of friction; in other words, the coefficient of friction is determined by the nature of the lubricant and not by that of the rubbing surfaces, when the latter are in good order. Soft metal or lined bearings heat more than bronze, and run at temperatures somewhat higher with equally free or even freer feed. This, however, does not necessarily indicate a serious defect, but simply deficient conduc-

tivity. The value of the white alloys for bearings lies mainly in their ready reduction to a smooth surface after any local or general injury by alteration of either surface or form.

SPECIFICATIONS FOR SLEEVE BEARINGS.

Based on the foregoing information, it is believed that specifications should provide for babbitt lined bearings for all motors of ten H. P. and over. For all motors under ten H. P. bronze bearings should be required. The reason for permitting bronze bearings on smaller motors is based on the fact that on smaller loads, bronze will give more service with less attention to lubrication. Under the same lack of lubrication babbitt would melt or wear out rapidly. On larger motors, bronze bearings, while satisfactory initially on an unworn shaft due to the hardness of the bronze, it is practically impossible on a worn shaft to produce a proper bearing fit on a replacement. Under this same condition babbitt will readily conform to the shape and form of the shaft.

It is believed that in order to produce more rugged and generous designs a formula could be used such as $P \times V$, shall not exceed 200, where "P" is the pressure in pounds per square inch of projected area (*i.e.*, product of diameter of the journal by its length), and "V" is the peripheral speed of the journal in feet per second. As a further safeguard to secure rigidity of the shaft under external load, the length of the journal should not be less than twice the diameter of the shaft.

EFFECT OF SPEED ON BEARING HEAT.

In order to dispel the popular illusion that bearing pressure is the chief determining factor in producing bearing heat, it will be of considerable interest to operators and designers to study the recently conducted experiments of Mr. E. G. Gilson at the General Electric Research Laboratory. In these experiments, the bearing pressure was reduced to zero by mechanically supporting the shaft on external bearings. The running clear-

ance was .015 inch, and the metal used was babbitt running on steel. The clearance space between the shaft and the journal was filled with mineral lubricating oil, and the temperatures were indicated by thermo-couples projecting into the bearing. Extremely high temperatures were evolved in a short time. At a surface speed of 9000 feet per minute, a temperature of 190 degrees C. was reached in 17 minutes. At a surface speed of 11,000 feet per minute, 190 degrees C. was reached in 8 minutes.

These experiments clearly indicate that the internal friction of the oil itself is, at the higher speeds, sufficient to burn out soft metal bearings and to entirely destroy the lubricating properties of the oil. Further experiments were made with various combinations of metals and with various types of lubricating oils. High temperatures of variable amounts were evolved in every case, the variation depending on the characteristics of the oils and the comparative heat conductivity of the oils and metals used.

From the above, it should be borne in mind that, although heat is produced by bearing pressures, there is a definite limit to the speed of an oil lubricated sleeve bearing due to the additional heat produced by internal friction of the oil. When a properly designed, fitted and lubricated bearing runs too hot, it can be safely assumed that the speed of the shaft is too great. The heat effect of speed, likewise, shows on ball bearings. For the same reasons, a definite limit where grease lubrication should be changed to a restricted quantity of oil lubrication, is indicated under paragraph on lubrication of ball bearings.

ROLLER BEARINGS.

Roller bearings being closely associated with ball bearings as a substitute for sleeve bearings, it is desirable to define the usefulness and limitation of roller bearings as applied to motors. The original idea was to combine pure rolling motion with great capacity. Natural laws, however, defeat the purpose. On a cylindrical roller bearing, if the rollers are tapered slightly

from mathematically true, or are permitted by looseness of design or assembly, to deviate from parallism with the axis of rotation, there is slipping and grinding, the area of the contact is unknown, variable and indefinite. On tapered roller bearings if the angle of the races is not mathematically correct, and the angle and size of the rollers are not mathematically correct to correspond, the deficiencies will result in slipping and grinding. The law of true rolling motion is stated, that, "when the axes of two bodies are at an angle to each other, the axes must meet in a point, and the lines of the surfaces must meet in the same point."

In realization of some defects of roller bearings, men sought to evade the law by making tapered roller bearings adjustable. Assuming that the angles and sizes were somewhere near right when the bearing was new, any attempt to compensate for wear results in the line of the outer race falling farther and farther inside the true meeting place, as the bearing wears and is adjusted. Grinding and sliding due to misalignment become progressively worse.

The tendency of the rollers to twist and wedge in the races cannot be avoided. Sometimes actual locking or breakage is caused in this manner. On tapered rollers there is a resultant pressure force, tending to slide the roller endwise. This force, in conjunction with centrifugal force which is large at high speeds, is restrained by the use of a shoulder of some kind in all taper roller bearings.

Roller bearings came into use because of the poorly designed and wrongly applied earlier type of ball bearings, which would not "stand up" to heavy loads for two reasons: First being poorly designed, the second, because the bearings were too small. Ball bearings run much easier than rollers, but good service cannot be expected of wrong design or application. The fact that a thing is defective does not necessarily mean that it will not run. It is hard to make a bearing so poorly that it will not run. It is the quality and kind of service that alone is a measure of value.

Tapered roller bearings, meant for both radial and thrust loads, are very inefficient thrust carriers. The angle commonly used indicates that the resultant pressure per roller under thrust is very high, and therefore the resistance to revolving must be proportionately high.

A roller bearing made as well as the inherent defects of rollers permit, is at least preferable to a poorly designed and wrongly applied ball bearing, for speeds practical for rollers. The above remark, therefore, may explain the prevailing success which taper roller bearings have met with in the automobile industry in comparison with ball bearings. As is well known, the better makes of automobiles use tapered roller bearings in the four wheel bearings, in the transmissions, propeller shaft and differential gear support. This success in its analysis to motor bearings must be, in the light of the fundamental defects of roller bearing, attributed to the comparatively low speeds at which the bearings operate, the very low continuous end thrusts or very intermittent high end thrusts under operating conditions, and the fortunate factor of general high class design and workmanship of the manufacturers who supply the trade in this class of material. This is in contrast with the numerous make-shift concerns who supply cheap and inferior balls and ball bearings of incorrect design.

BALL BEARINGS.

The question as to whether ball bearings should be used on motors is tied in with two phases of the bearing situation; namely (1) Are the disadvantages of sleeve bearings serious enough to warrant the substitution of ball bearings? (2) Do the advantages of ball bearings outweigh the disadvantages of ball bearings? As an incomplete answer to the problem, it can be stated at this stage, that without a thorough understanding of ball bearings, even if theoretically superior to sleeve bearings, they become practically less desirable. This is for the reasons that the general design, application, diameters of balls, diameters of inner and outer races, width of bearings, selection

of radial, angular and thrust bearings, method of locking the races, allowances for shaft expansion, lubrication, housing seals, specifications for materials, ball spacers, adjustable and non-adjustable bearings, etc., all are not practically understood by the motor manufacturers.

As there can be no doubt in the minds of engineers who have studied the problem of bearings, that ball bearings have definitely entered the field, the remainder of this article will be devoted to information which will, firstly, guard against the use of inferior ball bearings and improper use of superior ball bearings, and secondly, which will point the way to the proper selection and application of suitable ball bearings.

ADVANTAGES OF BALL BEARINGS.

(*a*) Over-all length of machine is reduced.

(*b*) Suitable for very cold temperatures where oil sleeve bearings would fail through insufficient lubrication.

(*c*) End thrust can be readily absorbed without undue heating or wear.

(*d*) Floating end thrust can be reduced to a minimum by virtue of the fact that ball bearings can be designed to continuously carry end thrust without undue wear or heat.

(*e*) As no oil rings are required for lubrication, all the disadvantages of oil rings are eliminated.

(*f*) Ball bearings will continue to run in emergencies with emery or other serious foreign material accidentally or surreptitiously placed in the bearing.

(*g*) By saving in space, the load on the driving end can be applied nearer to the center of the bearing support with consequent less deflection of the shaft.

(*h*) Although oil lubrication is superior to grease, the fact that ball bearings will operate efficiently at the usual motor speeds with grease as the lubricant, eliminates all the inherent excess oil troubles which sleeve oil lubricated bearings experience.

(*i*) The mechanical friction of ball bearings being from 30 per cent to 60 per cent less than the friction of sleeve bearings, results in saving of power input and reduces also the temperature rises of the motor. The saving in power costs at commercial rates would pay for the additional cost of ball bearings in a short time.

(*j*) As the static friction is very high for sleeve bearings, and is very low for ball bearings, the use of ball bearings results in considerable saving in power for motors which are frequently started and stopped.

(*k*) Very high shaft speeds can be practically obtained by the use of oil-fed ball bearings. This permits designers to consider higher speeds than are practicable for sleeve bearings.

DISADVANTAGES OF BALL BEARINGS.

(*a*) It is difficult to determine the quality of the material on every individual ball bearing prior to assembly in motor. On properly designed and applied ball bearings, their success depends on the individual quality.

(*b*) Practically no part of the ball bearing, including balls, ball races or ball retainers, can be fabricated on shipboard.

(*c*) The shaft and housing must be of fairly accurate diameters to permit proper fit of ball bearings.

(*d*) Worn shafts and housings, if trued up, would be under and over size for the original size bearing. This condition should only occur when the original bearing is too loose for the motor.

(*e*) Standardization of bearing diameters and widths, to the least number of sizes, is necessary for economy in spare parts.

(*f*) The cost of well designed and fabricated ball bearings is considerably higher than that of sleeve bearings.

(*g*) Salt water, moisture, acidity in oil or grease seriously corrodes or pits the balls and ball races, with resultant destruction of the bearing.

(*h*) The failure of one ball in a bearing, from any cause, requires as good practice that all the remaining balls be also

replaced. This is for the reason that all balls will gradually wear under size. The new ball of proper size is accurate to .0001 inch diameter and, therefore, cannot be properly assembled with the used balls which may be worn .001 to .005 under size, for the reason that the new ball would be overloaded due to its excess diameter over that of the old balls. For the above reasons, it is a disadvantage, as many users make a practice of replacing only a few balls in bearings which may be broken or pitted.

(*i*) Many ball bearings are made which cannot readily be disassembled for replacement of the balls. It is usual to discard the entire bearing for a defect in one small part. This causes an economic loss.

(*j*) On non-adjustable bearings, which are designed to permit ready removal of the balls, the ball race is usually weakened by such a design.

(*k*) Difficulty is experienced in the efficient design of ball spacers. On efficiently designed and fabricated ball bearings the ball spacers are considered to be a serious part of the successful bearing. While little lubrication is required for the balls rolling in the races, efficient and continuous lubrication is required for the ball spacer.

TYPES OF BALL BEARINGS.

For the purpose of narrowing the subject to practical application to motors, ball bearings may be divided into two general classes: (*a*) Adjustable. (*b*) Non-adjustable. In order to clear the way for the adoption of only the non-adjustable type of ball bearings the following inherent characteristics of adjustable ball bearings are set forth.

ADJUSTABLE BALL BEARINGS.

When it is considered that the safe compression of a ball under load is measured in tenths of thousandths of an inch, it is apparent that any adjustment made with a screw driver or a

wrench must necessarily be relatively crude. If a sleeve bearing is adjusted too tight it "freezes" or burns. A ball bearing gives no such warning. It may be seriously overloaded by tight adjustment, yet appear to run quite freely, or it may be loose without indicating its looseness by visible or audible signs. If loose, one ball at a time may take the whole load. If tight, the races and all of the balls are overloaded and distorted. In either case, premature failure results.

Moreover, the wear due to maladjustment or normal operation is constantly progressive, while hand adjustment, crude as it is, is only occasional. Attempts have been made to make constant adjustment by using external thrust springs. In such construction the spring pressure must be greater than the greatest thrust load. The causes of wear, internal grinding, and wedging, are not in the least reduced, but actually increased by the continual heavy spring pressure. Therefore, it may be said that any external adjustment is, in all probability, a prolific cause of trouble, and that any hand adjustment is at the mercy of the most incompetent operator.

TYPES OF NON-ADJUSTABLE ANNULAR BALL BEARINGS.

Annular ball bearing, meaning ball bearing in the form concentric with respect to the shaft, are classified into two general types: (*a*) radial; (*b*) angular. For reasons previously given, adjustable bearings will be excluded and further description will be narrowed to the non-adjustable types.

RADIAL BALL BEARINGS.

The true form of radial bearing is one in which the load is supported at right angles to the shaft. Under such conditions the bearing should have a correct rolling motion without wedging, slipping, or spinning of the balls on their axes. In the modern bearing of this type the working surface of both the inner and the outer races are curved in the direction of the longitudinal axis of the shaft and with centers lying on radii of the shaft. The radius of this curvature is always greater

than the radius of the balls. Thrust loads cannot be taken on a purely radial type bearing. Under thrust, it is distorted and becomes a very poor angular contact bearing. This produces a wedging or bursting effect of the balls on the races which may be many times as much as the end thrust applied. Slight radial looseness on this type of bearing gives excessive end play.

As radial bearings of the type described give a true rolling motion under radial loads, such as supporting a motor armature, they can be correctly applied to at least one end of a motor shaft providing the opposite end has a bearing which will correctly take a thrust load in either direction.

ANGULAR BALL BEARINGS.

In order to meet the requirements of end thrust in addition to radial load, the angular ball bearing was evolved. This is supposed to cover the requirements of radial load, and thrust load in one direction. With a double ball race, in which one-half is reversed with respect to the other half, end thrust in both directions is taken up.

There are two fundamental types of angular ball bearings in use. The first, being the most usual, consists of having the inner and outer curved ball races designed at an angle so that on a cross-section the tangents would be about 30 degrees to the shaft. Therefore, it can be estimated that the safe radial load would be twice that of the end thrust load. This type is known as the two-point contact type.

The second type, which is not so well known, is designed so that on the cross-section the tangent of the inner curved race is at about 20 degrees and the outer curved race at about 40 degrees to the shaft. As these tangents are not parallel, the balls tend to move out in the open angle between races under radial, or thrust load in one direction, but is prevented from so doing until the forces are balanced against an annular flat sided ring which is free to rotate with the balls. This is known as a three-point contact bearing, although it should not be con-

fused with other types of three-point contact bearings of incorrect theory.

The greatest disagreement arises amongst the various manufacturers concerning the fundamentals of the design and theory of these two types of angular bearing in regard to the purity of the rolling properties under radial and thrust loads at all speeds. One manufacturer convincingly illustrates, by theory and test, that the two-point contact angular bearing will not take care of either radial or end thrust without wedging, grinding and spinning of the balls. If this assertion is accepted, herein lies the secret of the unsatisfactory service of this type of bearing, especially at the higher speeds where the inherent defects are more pronounced and destructive.

There is an agreement that the purely radial type bearing, as previously described, will not take care of end thrusts. This reason must be on account of wedging and grinding of the balls on the races. On the two-point contact angular bearing, the tangents of the inner and outer races being parallel at rest, would properly be analyzed as likewise having a wedging and grinding effect under radial or thrust loads at all speeds. Such bearings at high speeds would become centrifugal brakes. The same wedging effect in a static form is used as a popular rebound check for automobile springs. The fact that such bearings actually run merely proves that the defects are not sufficient to cause immediate failure, although the friction and wear are present.

BALL SEPARATORS.

Ball separators were originally ball retainers designed to merely prevent losing the balls on disassembly. From a study of the failures of perfectly good balls when used without separators, it was learned from theory and practice, that the separator has an important function other than that originally contemplated. The primary function of ball separators is to prevent the balls from crowding, with consequent grinding. The theory of the crowding is as follows: under a load ver-

tically down on a radial bearing, the top ball gets the heaviest load, and, therefore, has greater compression. Hence the effective diameter of the ball under greatest load is less than that of adjacent ball and advances proportionately slower. The ball in back of it, as it advances, is slowing down as it approaches the same condition. The ball ahead is speeding up by an equal amount as it leaves this condition of compression. The actual amount is extremely small, but is sufficient to cause heavy pressure between adjacent balls if they are permitted to be in contact. As their surfaces are going in opposite directions at the point of contact, destructive grinding results. Hence the need of ball separators is a recognized factor in correct design.

Due to the important function of ball separators as above described, it is essential that they should be well designed and of proper material, or otherwise they wear out and break, thereby causing destruction of the bearing. As separators travel with the balls, they should be stiff enough to stand the centrifugal forces, and should be in running balance.

The separator being in pure sliding contact with the balls, there should be ample surface of contact such that the oil film would not be broken through ball pressure on the separator. The metal of the separator should have good anti-friction properties, high heat conductivity and elasticity. In addition, the separator should occupy as little space between balls as possible consistent with ruggedness, so that the maximum number of balls can be applied. Screws and rivets are to be avoided, as many failures have resulted in these working out into the ball race. On double row ball bearings, it is essential that the separator be in two units and not in one piece construction for both rows. This is for the reason that in the one piece separator, unless the balls and races have mathematically the same diameter and compression, one row of balls will tend to revolve at a different rate of speed than the other. Being restrained by the one piece separator, there results additional friction and wear.

LUBRICATION OF BALL BEARINGS.

The greater reliability of the ball bearing results partly from its less exacting lubrication requirements. Sleeve bearings must have a comparatively large amount of oil correctly applied to maintain the essential oil film. With ball bearings, lubrication is not so important, as it is not directly connected with the load carrying capacity. For this reason they will continue to operate even if the supply of lubricant fails temporarily. Thus, the use of ball bearings removes the need for constant watchfulness required of sleeve bearings. They are, therefore, valuable for installations in inaccessible locations.

There exists considerable diversity of opinion among the users and manufacturers of ball bearing motors as to whether oil or grease should be used as the lubricant. It may be said, however, based on very sound knowledge of lubrication, that oil as a lubricant is superior to grease.

With the above principle of lubrication in mind the problem of the selection of oil or grease as a lubricant remains purely one of convenience and practicability. The following characteristics of grease and oil lubrication are presented as a guide to designers and operators:

(*a*) Grease will form a more effective seal against entrance of foreign matter into the bearing as well as to cause less trouble from leakage.

(*b*) When abrasive matter has once gained entrance into the housing it is generally more destructive to the bearing with grease than when oil is used. Grease tends to retain the foreign matter and present it over and over again to the bearing surface, while oil allows a certain amount of segregation due to settling.

(*c*) With oil lubricated bearing, abrasive matter may be readily flushed out without removing the bearing. With grease this is only partly possible.

(*d*) Grease should be used with a melting point of at least 40 degrees to 60 degrees F. in excess of temperature of adjacent parts.

(*e*) Grease is suitable only for low speed bearings, as at high speeds it looses its plastic state and becomes a fluid of but poor lubricating properties. The bearing, not being designed for fluids, would present mechanical difficulties to their use.

(*f*) At high speeds, 5000 to 10,000 R. P. M., and very high speeds (in excess of 10,000 R. P. M.), it becomes necessary to use oil. In such cases the problem is one of limiting the quantity of lubricant in the bearing so as to prevent heating from fluid friction. Some form of sight feed or wick feed is generally used for this purpose.

(*g*) Only a small amount of grease is necessary in a bearing. Forcing an excessive amount into the bearing by means of a compression grease cup, or other device, leads to heating and often results in melting and breaking down the grease, so that its lubricating value is destroyed. As a result, further heating takes place and if more grease is added it will still further aggravate the cause of the trouble. In such cases the oily or lubricating constituent will leak out from the housing, leaving the bearing entirely caked with the hard filler having no lubricating value whatever.

(*h*) When oil is used for lubrication it is desirable to use an oil well of ample capacity in which the oil level is maintained at a point where the lower balls are partially submerged. Thorough lubrication of all parts of the bearing is accomplished by the oil picked up by the submerged balls. The purpose of the oil well is to permit the settling of foreign particles of metal or grit, and to allow settling of the sludge from the oil produced by the bearing heat.

HOUSING SEALS.

The effectiveness of the housing seal bears an important relation to the lubrication of a ball bearing. When a rotating shaft extends through a stationary housing, it is necessary to provide some form of seal between these elements. The function of the seal is to:

(*a*) Prevent entrance of dirt and foreign matter into the housing.

(*b*) Prevent leakage of the lubricant from the housing.

The seal design is of fundamental importance in the construction of any ball bearing mounting. The usual type of seals include oil and grease grooves, felt washers, labyrinth seals, and oil and dirt flingers. In general, oil lubrication necessitates the use of a more elaborate type of seal than is necessary for grease. Great care must be used in the design of all-metal seals to permit large enough clearances for wear of bearings and flexture of shaft, and at the same time act as an efficient seal. With felt washers, the friction heat due to the felt rubbing on the shaft, must be recognized and provided for in the design, as a moderate pressure on the felt and high speed of the shaft will cause the felt to burn.

PRACTICAL EXPEDIENTS ON BALL BEARINGS.

The successful operation of ball bearings sometimes is prevented by the lack of knowledge of mechanical expedients necessary for operating conditions. Some of the important ones are as follows:

(*a*) It is essential to bore the housing one thousandth of an inch per inch of diameter longer than the bearing, up to .005. Otherwise the bearing will bind from the unequal expansion of the bearing and the housing.

(*b*) Narrow bearings peen the shaft and housing, to offset which many false expedients have been used, including press and shrink fits. This is not only mechanically wrong, but inconvenient if not disastrous in replacement or disassembly for cleaning and inspection. The solution is to have the bearing wide enough and fit properly so that a slip or push fit on the shaft or housing produces the desired results.

(*c*) The shaft of a motor armature expands in length due to the temperature of the operating parts of the motor. This expansion is serious in amount compared with the working limits of ball bearing fixed adjustments. Compensation for

the shaft expansion must be provided for by having the bearing on one end, preferably the end opposite the driving end, free to float endwise. This is accomplished by having the outer race a revolving fit in the housing, and with an end clearance in the housing of sufficient amount to take care of the minimum and maximum temperature conditions.

(*d*) A ball bearing motor, when directly connected to a machine which has sleeve bearing or reciprocating parts, should be provided with a flexible coupling. This is for the reason that sleeve bearings wear appreciably, whereas ball bearings do not; misalignment would therefore result if a flexible connection were not used. The flexible coupling would relieve the ball bearings from the shock of reciprocating parts.

SPECIFICATIONS FOR ANNULAR BALL BEARINGS.

From the foregoing paragraphs it is apparent that the only safeguard in the purchase of suitable ball bearings is by the adoption of rigid detailed specifications. The high reputation of well known manufacturers cannot be depended upon entirely to safeguard the interests of the Navy. There must be, under present conditions, considerable necessity to obtain ball bearings as replacements for bearings which have come to a premature end due to poor design and materials. This, of course, is in addition to replacements caused by improper application, abuse or lack of attention. As a tentative specification, the following requirements are considered essential:

(*a*) Bearings shall be of the annular ball bearing type, non-adjustable.

(*b*) The radial looseness shall not exceed .0003 inch, and shall spin freely in the fingers to indicate that the balls and races are not too tightly assembled.

(*c*) Inner races shall be a minimum of one shaft diameter in width.

(*d*) Bore and diameters shall conform to the sizes and limits of the Society of Automotive Engineers.

(*e*) Eccentricity with inner race revolving and outer race stationary shall not exceed .001 inch.

(*f*) Material for balls and races shall be chrome alloy steel, with no nickel, of the following analysis: Chromium 1.25 to 1.50 per cent, carbon .85 to 1.00 per cent, prosphorus and sulphur .0025 per cent maximum, silicon, manganese and other beneficial elements allowed according to the state of the art.

(*g*) Case hardened, carbonized or soft steel not permitted in construction of balls or races.

(*h*) The balls and races shall be structurally hard equivalent to 75 to 85 Brinell. The degree of surface hardness shall be such as to be just "touched" with a fine Swiss file. Upon fracture they shall show a fine even grain, without evidence of overheating, slag inclusions or other weaknesses.

(*i*) Ball races shall be lapped. No grinding marks, waves, chatter marks or pits will be permitted in the actual ball track. This examination will be made with a strong magnifying glass or low power microscope.

(*j*) Balls shall have a high polish, free of surface defects, shall be round and true to size within .0001 inch. Balls in different bearing will be permitted a variation in size to .0004 inch, but all balls in the same bearing shall be within the limits of .0001 inch.

(*k*) All bearings shall have ball separators for circumferential spacing of the balls. The separator must run concentric within .020 inch. If of stamped, pressed or sheet metal, edge contact with the balls will not be permitted, but the separator shall be formed with pockets substantially fitting the surface of the ball for the width of the separator. Material, if sheet, stamped or pressed metal, shall not be less than 1/32 inch thick for bearing under 1⅜ inch bore, and not less than 1/16 inch for bearing over 1⅜ inch bore.

METHOD OF TEST OF BALL BEARINGS.

The old conventional method of overload test at low speeds results possibly in the selection of the poorest bearings. To

stand excessive overload it is only necessary to have the balls nearly fit the raceways by having the diameter of the raceways just slightly larger than the diameter of the balls. Excessive overload tests therefore, mean nothing except the possibility of the determination of ultimate limits. Bearings fail in service from fatigue and wear. Speed increases these factors but does not change the inherent design tendency to fatigue and wear. To reduce the time element of test, high speed is used as the short-time endurance test, using either normal load or no load.

It is considered, as a comparative test of different designs of ball bearings, that a good bearing should run 500 consecutive hours, with flood lubrication, at a ball pitch line speed of about 3000 feet per minute, without developing excessive noise, wear or looseness.

The above test which may carry the shaft speed up as high as 18,000 R. P. M., was evolved to eliminate the improperly designed load carrier in which the balls merely fit the races. If the run is made at normal load, with a suitable proportion of thrust, the high speed test will likewise eliminate the incorrect designs in a few hours as evidenced by excessive wear.

The statement is sometimes made that the capacity of a ball bearing decreases as the speed increases; this is not sound. There appears to be no theory connected with the statement, and it is probably only the characteristics of a poorly designed bearing in which the inherent wear is retarded by decreased capacity proportionately to the accelerated wear produced by increased speed.

PROPER TYPE OF BALL BEARINGS FOR MOTORS.

From analysis of the characteristics of the various types of ball bearings it is concluded that the following combination and types of bearing are the best suited for motors: The commutator end of the motor to be provided with a single row radial ball bearing with the inner race secured to a shoulder on the shaft by means of a clamp nut with means for locking the nut and with the outer race permitted to float end-wise to

allow for shaft expansion. The driving end of the motor to be provided with a double row angular ball bearing, designed to correctly take both radial and all thrust loads. The inner race to be secured to a shoulder on the shaft by a clamp nut with means for locking the nut. The outer race to be securely clamped in the housing by bolted housing cap.

BALANCING L. P. TURBINE ROTOR ON BOARD U. S. S. *DETROIT*.

By Comdr. H. C. Dinger, U. S. Navy, Member.
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I have read with great interest Commander Bean's article, "Method of Balancing Turbine Rotors of the Scout Cruiser *Raleigh*," and a discussion of it by Captain P. B. Dungan, in the May number.

The device described is certainly an interesting and ingenious one for obtaining the approximate balance of a rotor by spinning up to speed, which method would be used in cases where no balancing machine is available. The method used, however, appears to be an approximation. If a balancing machine is available, as, for instance, the Akimoff machine, a rotor can be balanced more quickly and more accurately than by the method described. Such balancing would usually be done in the shop shortly before the rotor is taken to the vessel for installation.

Akimoff balancing machines have been in use in various Navy Yards for the past 5 years, and have been found very useful for balancing turbine rotors, armatures, shafts, blower fans, etc. The balancing is usually done in the shop, although the machines can be set up almost anywhere.

By employing the balancing machine the amount and angle of unbalance can be quickly determined, and from this data the exact position of the correcting weights can be determined. The resultant unbalance, due to static and dynamic unbalance, can be determined and corrected in one operation, and there is no special need of correcting the static and dynamic balance

separately. The static and dynamic unbalance can be determined separately if this is for any reason desirable. With ordinary units of symmetrical design the process of balancing becomes very simple and requires comparatively little time. With very large units of unsymmetrical design the balancing procedure may be considerably lengthened, but during the past 5 years at the Navy Yard, Philadelphia, no balancing problem has been met which has not been solved in a reasonable time.

The parts of the Akimoff balancing machine are portable, and can be moved about readily, to suit varied locations and conditions. The largest single piece involved in the apparatus is the motor. Recently at the Philadelphia Yard a low pressure rotor of the U. S. S. *Detroit* was completely balanced on the ship. It is believed that this was the first time that the main rotor for a vessel was balanced by using a special balancing machine without removing such rotor from the vessel.

On January 25th, 1924, while the upper casing of No. 1 L. P. turbine was being lowered to close up the turbine after straightening bent blades and securing some loose shrouding, a chain-fall failed and the casing was dropped. This accident resulted in damaging the blading and breaking two guide lugs on the lower casing. It was necessary to replace about 600 blades.

Immediate dynamic rebalancing was considered essential. As the engine room hatch of this vessel is closed in, removal of the rotor to the shop would have involved a great amount of work. The rotor was examined, spun in place and observations made with dial indicator to establish the fact that the spindle had not been damaged by the accident and that the rotor ran true. It was then decided to set up the balancing machine in the engine room and balance in place.

The ship was floated in dry dock so as to be held free from oscillations due to local disturbances on the water front. Some of the final balancing was done at a pier on the Delaware water front and no difficulty was experienced, it is considered,

however, that the ship should be in still water and on even keel and unvarying trim until unbalance has been practically corrected.

The upper casing was hung off and blocked at each corner with 12-inch × 12-inch timbers. (See Figures 1 and 2.) Supports to carry the balancing machine and rotor were built up from the lower casing with 10-inch I-beams and 10-inch channels as shown in Figure 1. Thwartship and diagonal braces, 3-inch × 3-inch angle bar from the back of channel to the turbine casing, were installed, and are shown in Figure 2. In this Figure there also appears the preventer thrust arrangement and wire preventer strap which is held clear of the spindle and intended to support the rotor in case it should roll out its bearings. The regular Akimoff thrust rod was mounted on a similar thrust bracket at the after end.

The motive power was supplied by a 5 H. P. variable speed motor, 500-1000 R. P. M., located outboard of the turbine on a large fore and aft girder at the level of the engine room upper grating. The driving belt was rove over the middle of the spindle with pulleys so arranged as to produce a vertical pull on the shaft and to reduce speed to 135-270 R. P. M. This location for the drive differs from shop practice of driving from the fixed end and, at first, there was some doubt about the effect of the belt tension in damping vibrations. Results indicated that there was no appreciable effect due to the belt. A mechanical hand brake of nut cracker type was used to bring the rotor to rest as a substitute for the electrical motor brake which is used in the shop. This brake consisted of two yellow pine sticks shaped to fit the shaft and held together by a leather strap as a hinge. Instead of turning the rotor end for end after balancing one end, as is done in the shop, the sensitive bearing and the fixed bearing with thrust rods were interchanged. The motor control rheostat was so placed as to be convenient to the sensitive end and was shifted to the other end when the bearings were shifted.

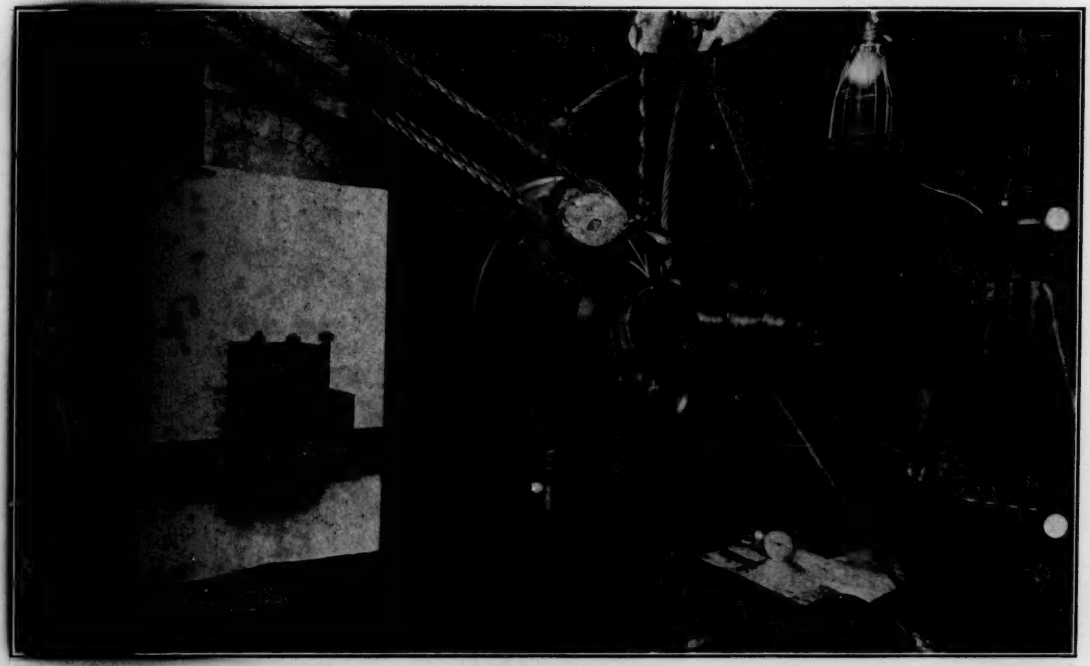


FIG. 1.

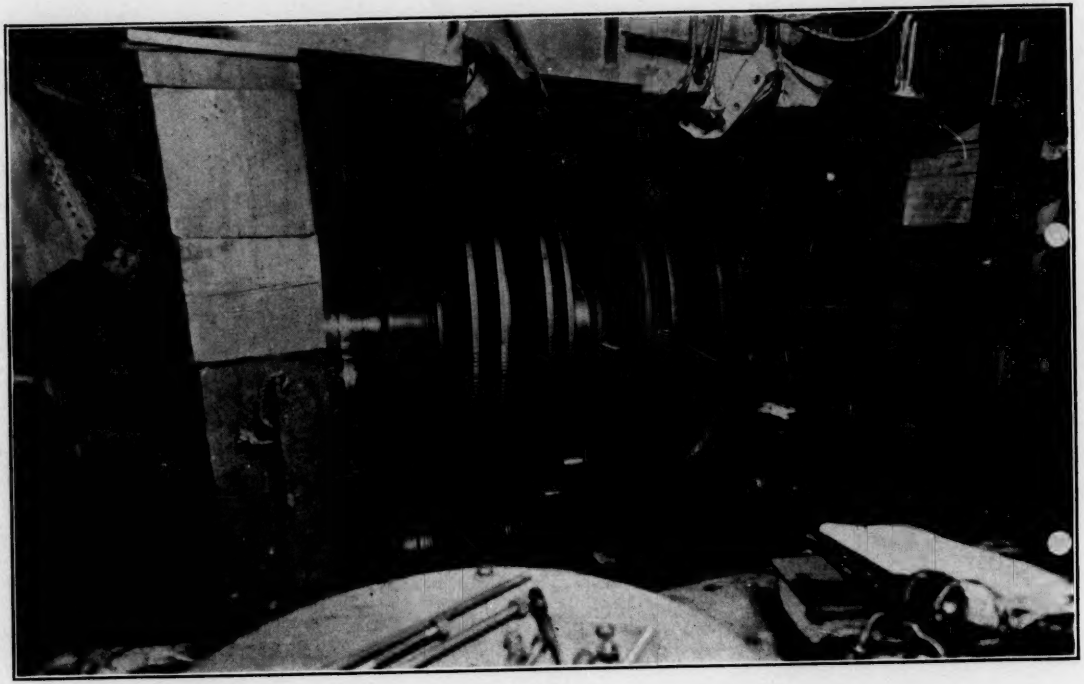


FIG. 2.

At the outset of the balancing operation it was at first necessary to introduce considerable damping in order to bring the vibrations down to safe limits and within scale readings. On account of the cramped space it was necessary to shorten the pendulum arms about 14 inches. With this length of pendulum (after all damping had been relieved) a critical speed was found at 215 R. P. M. From this point the behavior of the rotor and the procedure followed were in accordance with usual shop practice, except that on account of the unusual weight required to correct unbalance and on account of high centrifugal stresses resulting from a speed of 2500 R. P. M., the corrections were distributed in ten weights. A preliminary balance was made using four pairs of weights. On trial there remained a comparatively small unbalance which was corrected by the addition of two single weights.

Upon final spinning with the ten weights installed the maximum vibration was indicated by scale readings of .002 on either side of zero, the dial indicator being located at the extremity of the No. 4 balancing machine as shown in Figure 1.

After closing the turbine and before coupling the rotor to the reduction gear, the turbine was spun in both directions with steam, up to 2500 R. P. M., with no appreciable vibration at any speed.

It is believed that the use of solid cast iron blocks, or large oak blocks, steel faced, instead of the I-beams, channels and braces, would be more satisfactory as they would be more rigid and much easier to set up. The rigging finally used on this job was apparently sufficiently rigid to obtain a good balance, but without the braces the supports showed considerable thwartship movement. It is thought that one pair of the blocks furnished with the Akimoff machine would answer the purpose. They are 24 inches deep, which is only 4 inches more than the total height of the supports used.

If two sensitive bearings of the same size were obtained, the work would be simplified in that it would be unnecessary

FIG. 2.

to shift the bearings in changing from forward to after end. The bearing that should be stationary can be made so by setting up on the stops provided for limiting the vibration.

In comparing time and costs of this procedure with shop balancing it may be said that in preparing and setting up the ship apparatus ready to spin, approximately 6 eight-hour days were used. Of these 6 days, 3 were prior to the time that the reblading of the rotor was completed. In interchanging the bearings end for end, approximately 6 hours were used. The material cost was very little, as the principal items, including the motor and rheostat, were on hand in the shop. Breaking down the apparatus required about four hours. The costs and time of actual balancing, excluding the preparation and shifting described above, were practically the same as they would have been in the shop. By the expenditure of about 7½ days, and roughly $400 for direct labor, the removal of decks and interferences to permit the landing and replacing of upper casing and rotor, and transportation of the rotor to and from the shop were avoided. Of the total time required for the balancing about 10 days were chargeable to the abnormal unbalance of this rotor as compared to previous balancing of rotors of similar size which have been handled at this Yard. This abnormal unbalanced condition is believed due largely to the fact that the reblading of this injured rotor was done, in the main, on one side only, and thereby the selective feature of locating blades of approximately the same weight at intervals of 180 degrees from each other was not practicable in this case.

The above is an account of a balancing procedure under unusual conditions, and shows the possibilities to which balancing machines of the Akimoff type may be put.

As neither Captain Dungan nor Commander Bean refer to the use of specially designed balancing machines, it has occurred to me that the extent to which these machines have been used during the last few years may not be generally

known. The Akimoff machine is the one most generally used in marine work, but other types are also in more or less general use. The Kirkwood machine, handled by the Gisholt Machine Company, of Madison, Wis., is used extensively by manufacturers of automobile engines. It is understood that several other makes of balancing machines, in which the unbalance is actually weighed and the angle measured, are in use in this country. Experience in balancing with the use of the Akimoff machine indicates that it is often very desirable to have the rotor in approximate balance before being put on the machine. If the amount of unbalance is great, much more time is required to do the final balancing. It is also sometimes difficult to find good places to attach large weights. In some designs of turbines the clearance is small, and considerable ingenuity must be exercised to locate the weights so that they do not interfere with other matters. The most desirable way to correct unbalance is to take the necessary weight off the heavy side. This, however, can only be done to a limited extent.

A symmetrical unit, if it is in static balance, is usually not far out dynamically. While the operation of putting the unit first in approximate static balance may not be necessary, it is often very desirable in order to lessen the time for obtaining the final balance. This is so particularly where it is suspected that the unit is badly out of balance. It is sometimes better to take several steps rather than one big jump.

From my experience in the past six years, during which time hundreds of rotors of various kinds have come under my observation, I would say that any balancing or partial balancing is preferable to no balancing at all. Balancing on knife edges is useful, because it does eliminate a considerable part of the unbalance, and in a symmetrical unit, static balance will eliminate most of the vibration likely to be encountered, unless the unit revolves at very high speed. Complete balancing in an accurate balancing machine is, of course, the

most desirable procedure. This is especially so in cases where a rotor has been partially rebladed, and by this operation put out of balance extensively.

The matter of proper balance in rotating parts of marine machinery is a most important one, and merits much more attention than has been given to it in the past. If all rotors, shafting, etc., are accurately in balance, wear on bearings and the bad effects of vibration will be eliminated to a very great extent. Balancing is very important in parts rotating at high speeds (several thousand revolutions), but it is also desirable to balance those that rotate at moderate speeds. Balancing is not necessarily an expensive operation, and with a little experience on the part of the operatives, balancing can be handled about the same as any other shop operation. The future trouble that is avoided by proper balancing will pay for the extra care and cost involved many times over. All propeller line and propeller shafting should be balanced, at least statically, and these parts should be connected up with the turbine rotors so that the amount of unbalance in each individual piece is neutralized as far as possible by that in the others. If this is done, a total unbalance of the engine, shafting and propeller, considered as a unit, will be comparatively small. In fact it can be made almost negligible. Balance of rotating parts is very important in internal combustion engines and enables weights to be materially reduced.

DISCUSSION.

By S. Morris Lillie, Associate Member.

THE LILLIE EVAPORATORS ON U. S. NAVAL VESSELS IN VIEW OF A PAPER ON "LOW PRESSURE EVAPORATORS", BY LIEUTENANT EDWARD ELLSBERG, U. S. N., IN THE "JOURNAL", ISSUE OF AUGUST, 1924.

The paper of Lieutenant Ellsberg very severely criticizes the Lillie evaporators which have been installed on most of the major ships of the U. S. Navy. The deplorable conditions which the paper condemns, the difficulties in operation, and the failures in capacity stated are probably correct, for he speaks from his own experience, and he had had access to reports of operating experience of the personnel on the various ships he mentioned, and to Bureau of Engineering data; but the paper failed to state, that the evaporators referred to, were Lillie evaporator sets that had been modified by the Bureau of Engineering to a degree that made the modes of operation very different from the Lillie methods as practiced with normal Lillie evaporator sets, such as those on the U. S. S. *Dixie, Salem, Tennessee* and *California.*

Lieutenant Ellsberg was located at the Boston Navy Yard, and might not have been aware of the Bureau modifications. Beginning at the bottom of page 455, with the words: "Any person who has been associated with the operation of a Lillie type plant realizes the difficulties involved", etc., and on the following pages, he refers to troubles and difficulties in maintenance and operation which are not factors in the operation of a normal Lillie multiple effect. He speaks of very rapid falling off in capacity of the evaporator sets on the different ships. In the case of the Battleship *Maryland* he specifically states a fall in capacity which amounts to about 44 per cent in

one week, page 463, *i. e.* from 900 gallons per set to 500 gallons per set. On page 457 he speaks of the necessity of scaling the first effect coils weekly, and circulating dilute acid through all three effects for about eight hours.

He gives average outputs per set of a number of Battleships, as follows:

Utah—Two Double Effects. 800 gallons per hour per set.

Florida—Two double effects. 700 gallons per hour per set. Steam at 2 pounds pressure.

Maryland—Two triple effects. 900 gallons per hour per set, with steam at 4 pounds pressure; after one week's operation, due to scale, the capacity drops to 500 gallons at 5 pounds pressure.

Colorado—Two quadruple effects. 850 gallons per hour per set. Steam pressure 4 pounds.

These evaporator sets were all Bureau modified. Had they been connected up to form sets in a normal Lillie manner as on the ships *Dixie, Salem, Tennessee* and *California,* the productions would have been as follows:

Utah and *Florida*—Over 2000 gallons per hour per set, maintained for long periods without cleaning.

Maryland—Some 1400 gallons per hour per set, maintained for long periods without cleaning.

Colorado—Different reports up to 1800 gallons per hour.

In the case of each ship there would always have been a spare set as required by the rule laid down by the Chief of the Bureau. Either set could have cared for the needs of the ship by itself for an indefinite period. It is absolutely necessary to keep both of the Bureau modified sets on a ship in service to supply the ship's needs.

The falling down of the Bureau modified evaporators is due to scale principally, as stated in the Ellsberg paper. Information regarding the performance of many of the evaporator sets when first put in service on their respective ships, and before the Bureau modifications had had time to affect them, is that each set on its ship at the start showed production much

above ship needs, it would have been able by itself to supply the ship with distilled water for long periods, with the other set standing as a spare, had it been connected up as the Lillie evaporator sets on the ships *Dixie, Salem, Tennessee* and *California* were connected up.

The paper by Commanders Jones and Charlton entitled "Low Pressure Vacuum Evaporators (Multiple Effect Film Type)," printed in the May 1923 issue of the JOURNAL, opened with a description of the difficulties that had been had with evaporating plants, principally with respect to scale, and followed with these paragraphs:

"In 1912 an evaporator of an entirely new type was installed on the *Dixie*. Another plant of the same type was installed on the *Salem* in 1913. Although these plants were quite satisfactory and produced fresh water of excellent quality no further installations were made on Naval vessels until the *Tennessee* and *California* were commissioned in 1920 and 1921. The success of these plants together with the constant failure of other types of evaporators to operate satisfactorily led to the adoption in 1922 of this type of evaporator as a standard for Naval vessels.

"The adoption of this standard has resulted in specifying it for new construction as well as the conversion of existing plants, to operate on the new principles involved. The conversion is projected for all ships of the Navy as funds and time permit.

"The new standard type of evaporator is based on the patents of Mr. S. Morris Lillie *with modifications designed by the Bureau of Engineering* and will be known as the modified Lillie evaporator."

The statement regarding the Lillie evaporators on the *Dixie, Salem, Tennessee* and *California*:

"The success of these plants", etc. would never have been written had the evaporator sets on these ships had the failings stated in the Ellsberg paper for the Lillie evaporators (with

Bureau modifications); nor would the wholesale conversion of the evaporators on the Battleships of the Navy have been made. The records for these ships are as follows:

In the JOURNAL of the AMERICAN SOCIETY of NAVAL ENGINEERS, issue of February, 1913, the paper "Lillie Quadruple Effect Evaporator, U. S. S. *Dixie*" stated: "The capacity is only slightly reduced after long usage without cleaning."

After preparation for official acceptance tests before a Board, the quadruple effect was run at full rated capacity for a little over two weeks, after which it was subjected to a 72 days continuous run, and showed rating during the run, *i. e.,* from July 15th to 18th. The tests were continued under various conditions until the 3rd of August, and then showed rating operating with ½ pound steam to the hot effect.

Of the quadruple effect on the *Salem* there are no printed records. That its record was good is shown by the paper of Commanders Jones and Charlton.

For the *Tennessee* an instance is given of 45 days continuous operation without cleaning or scaling, plus a test of one week two days, or a total of 54 days. See paper "U. S. S. *Tennessee* Description and Official Tests," JOURNAL A. S. N. E., May 1922, by Commander C. A. Jones, U. S. N.

Of the *California* evaporators, which were duplicates of those on the *Tennessee,* we know of no printed statements of performance, but there are records of their running as two triple effects over 30 days continuously without scaling or cleaning, and maintaining their capacity at the end of the 30 days.

A notable instance was that of a Lillie octuple effect furnished the U. S. War Department for distilling sea water. It was guaranteed to preserve its rating throughout a 720-hours continuous run without cleaning. It was subjected by the Department, at its Sandy Hook Testing Grounds, to an acceptance test. At the end of the 720 hours continuous run the octuple effect was showing about the same rate as at the start, which was some 40 per cent above rating.

SEQUENCE OF LILLIE EVAPORATORS ON U. S. NAVAL VESSELS.

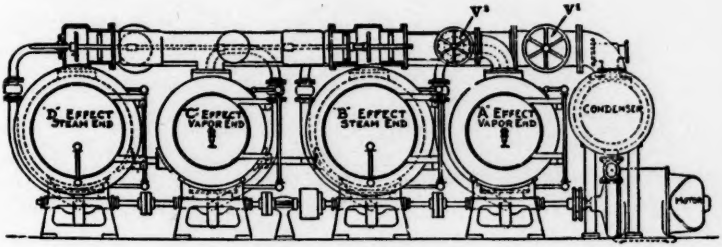


FIG. 1.—DIXIE TYPE.

On U. S. S. *Dixie*, 1912 to 1922, Then Transferred to the *Denabola*—*Dixie* Scrapped.

On U. S. S. *Salem*, 1912 Until Scrapping of *Salem*, 1921.

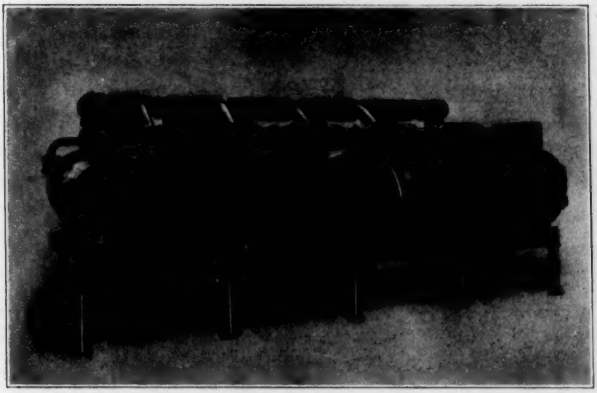


FIG. 2.—DIXIE TYPE.

On U. S. S. *Tennessee* and *California*, 1921 to Date. More Economical of Fuel than the Other Ships of the Pacific Fleet.

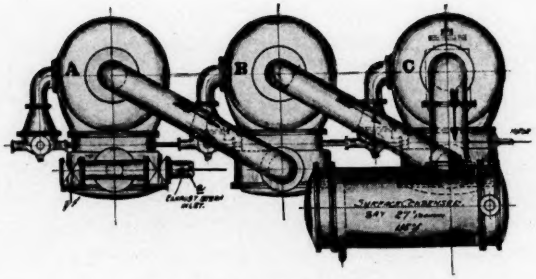


FIG. 3.—LILLIE 1922 MODEL.

128 Evaporator Units of Various Types Converted to This Form of the 1922 Model on Major Ships of the U. S. Navy. Patents Applied for.

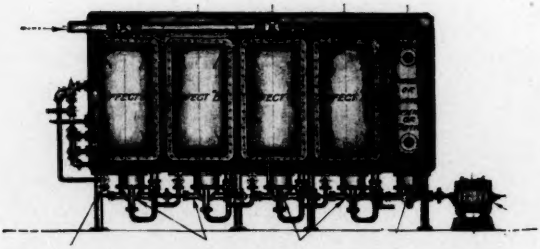


FIG. 4.—LILLIE 1922 MODEL MULTI-CELL CONSTRUCTION.

Eight Quadruple Effects, 32 Units Are Installed on or Are in Preparation for Major Ships of the Navy. Patents Applied for.

Two Lillie multi-cell triple effects were recently delivered to the War Department for Army Transport service. Each is guaranteed to maintain rating to the end of a 720 hours continuous run without cleaning when distilling sea water. A Lillie sea water multi-cell (five cells) quadruple effect and surface condenser is under erection in the Power House at Tocopilla, Chile, of the Chile Exploration Co.—the second installation of Lillie evaporators.

The Bureau modifications consisted largely in abandoning the methods of connecting up and operating which had proven effective on the said four ships in preventing the formation of scale, and in maintaining a constant output of distilled water for long periods. That these were done by the evaporators of the *Dixie, Salem, Tennessee* and *California* is implied in the paragraphs quoted above from the paper by Commanders Jones and Charlton.

I offered the Bureau of Engineering two new constructions of the Lillie evaporator, one to which the existing evaporators on the Battleships could be converted in situ, and the other a design for new construction evaporators. It would have been a very large undertaking to replace existing apparatus on the Battleships and other major ships with the Lillie evaporator as it was on the ships *Dixie, Salem, Tennessee* and *California;* but to convert the existing evaporators to the Lillie type in situ might be a comparatively small one. The Bureau was skeptical regarding the possibility of making these conversions, but was later converted by drawings and explanations submitted to it, in March, 1922, and then within 20 days the conversion of all evaporators on the Battleships and other major ships had been decided upon by the Bureau, and contracts were given as promptly as might be covering the entire 30 ships. This was a very remarkable proceeding by the Bureau of Engineering, which certainly would not have occurred had the Lillie evaporators on the *Dixie, Salem, Tennessee* and *California* been working in the inefficient manner of the evaporators as described in Lt. Ellsberg's paper.

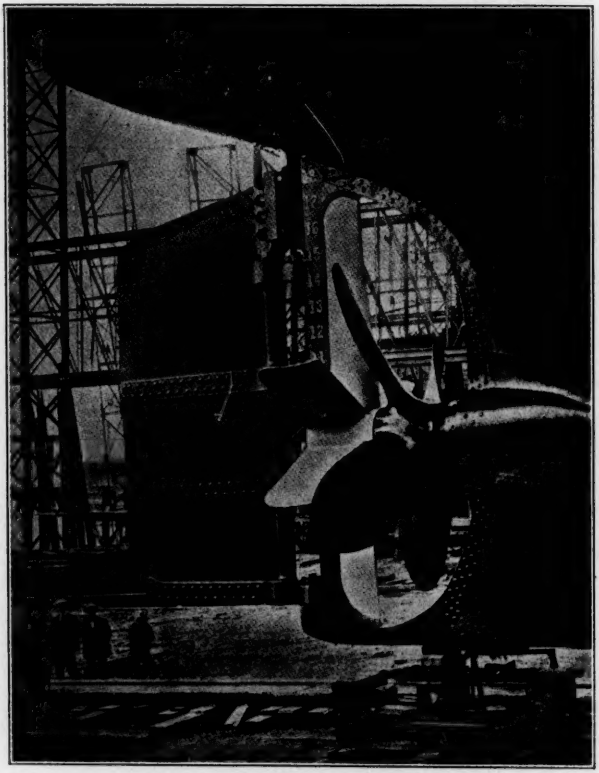
For new construction was offered the very unique multi-cell construction. Its advantages for naval service were so apparent that the day the drawings were submitted to the Bureau it was accepted for new construction work. The multi-cell construction was decided upon for Destroyers.

Evaporators of the multi-cell construction were installed with Bureau modifications on the ships *Colorado* and *West Virginia.* When the evaporators on the *Colorado* and *West Virginia* were first put in service, and before the Bureau modifications had had time to affect them, one set of the evaporators would produce at the rate of 1500 gallons distilled water per hour, which contrasts with the 850 gallons per hour credited to the *Colorado* at page 463 of the Ellsberg paper. This was after the Bureau modifications had gotten well into service.

The sequence of Lillie evaporators on U. S. Naval vessels is indicated in Figures 1, 2, 3 and 4.

The foregoing will indicate the injustice that is done to the Lillie evaporator by the Ellsberg paper not having contained some statement to the effect that the evaporators described are not normal Lillie evaporator sets, but Lillie evaporator sets with "modifications designed by the Bureau of Engineering."

It is suggested that all those who have read the paper by Lieutenant Ellsberg in the August issue of the JOURNAL will reconsider it in the light of the foregoing; and further, that all technical publications which have commented upon the said Ellsberg paper in their columns will make such corrections of their comments as may be thought advisable in justice to the Lillie evaporator.



STAR CONTRA-PROPELLER OF THE MOTORSHIP "SILVERLARCH."

THE WAKE OF A VESSEL FITTED WITH A STAR CONTRA-PROPELLER.

NOTES.

FIRST BRITISH MOTORSHIP WITH CONTRA-PROPELLER.

M. S. "Silverlarch" Fitted With a Star Contra-Propeller.

Much has been written lately with regard to devices which can be fitted to motorships in order to improve their general efficiency, and it must be confessed that more attention seems to have been paid to them in Germany than on this side of the North Sea—we refer to the Flettner rudder and to the contra-propeller. To the best of our knowledge, the first British motorship to be fitted with the latter device is the *Silverlarch*, which Swan, Hunter & Wigham Richardson Ltd. launched recently for the Mount Steamship Company Ltd., London, and which is a sister vessel to the *Silverpine*, which we described in detail in these columns in June. The *Silverlarch* is a three-deck vessel with a top-gallant forecastle and a poop, and she has the following dimensions and particulars:—

Length between perpendiculars, feet.....	400
Breadth, feet and inches.....	55—3
Depth to upper deck, feet and inches.....	28—8
Deadweight, tons	8,400
Draught at about, feet and inches.....	25—3¾

She is propelled by a single set of six-cylinder Neptune Diesel engines developing 2200 I.H.P. at 100 R.P.M. The contra-propeller, which is of the type constructed by the Star Contra-Propeller Ltd., Christiania, has been fitted to the ship in accordance with owners' wishes, and the welding in connection with the installation has been carried out by the Premier Electric Welding Co. Ltd., London and Newcastle.

There are several types of contra, or contrary-turning propellers, and the one in question has fixed blades attached to the stern post, one of the results of such an arrangement being, by getting the rotation out of the propeller race, considerable economy in propulsion, and experiments have shown, in the case of a tug fitted with a contra-propeller with fixed blades, that the thrust exerted by the screw was increased 50 per cent. In view of the fact that the *Silverlarch* is the first British exponent of the system, its performance will be watched with considerable interest. We illustrate the propeller, and we show also the wake of a vessel with a contra-propeller, the smoothness being very noticeable.—"The Marine Engineer and Motorship Builder," Aug. 1924.

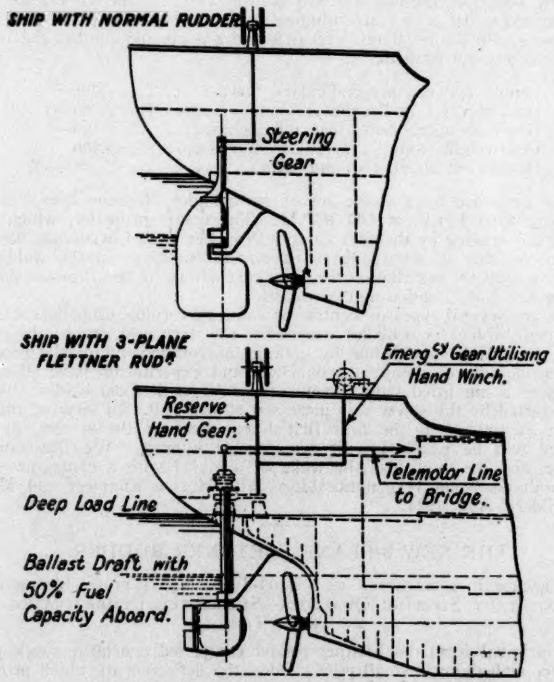
THE NEW 3-PLANE FLETTNER RUDDER.

First Practical Application in a 4500-Ton Dead Weight Motorship—Excellent Steering Qualities—Successful Demonstration on Trial Trip.

In its original form the Flettner rudder comprised one large single plane or surface embodying a small pilot rudder, the deflection of which provided a system of forces such that the large plane was caused to take up any desired position. The expenditure of power was so small for rudders of normal dimensions that hand power alone was adequate and steam or other

power-driven steering gear was unnecessary as only the small pilot rudder had to be operated. The practical success of the single plane rudders in the *Frigido* and *Odenwald* and the experience derived therefrom led to further experiments being made at the Hamburg experiment tank or Gesellschaft der Freunde und Förderer der Hamburgischen Schiffrau-Versuchsanstalt or, literally, the "Company of Friends and Promoters of the Hamburg Shipbuilding Experimenting Institute." The result of these experiments was the 3-plane rudder which, while embodying exactly the same principle of operation as the single plane rudder, gives a greatly increased steering effect.

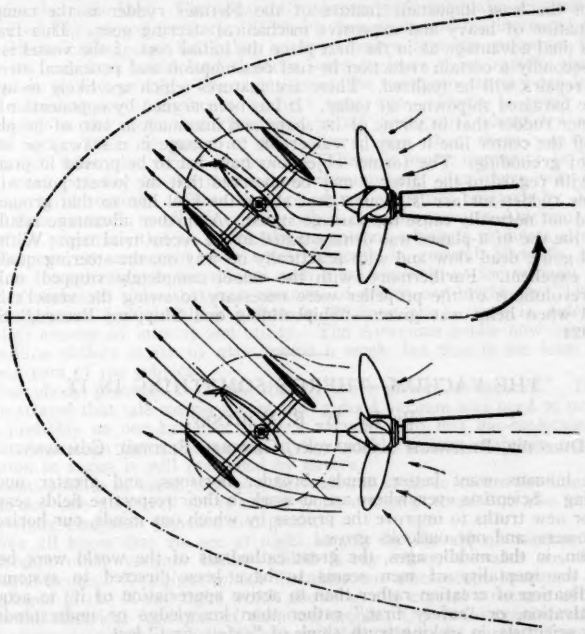
The system of multiple-surface rudders is not new and has been used in the largest warships and battle cruisers where, owing to the enormous weights and the great breadth of such vessels the single rudder was not adequate. In these ships, however, each rudder was placed at an independent point and all were operated by a common steering arrangement. The new Flettner rudder utilizes the system of multiple planes but these are closely disposed together and as in the earlier form of Flettner rudder the principle of the pilot or auxiliary rudder is incorporated in the center plane of the three. The center plane is connected to the rudder post or shaft by a flange


SHIP WITH NORMAL RUDDER
Steering Gear
SHIP WITH 3-PLANE FLETTNER RUDDER
Emergency Gear Utilising Hand Winch.
Reserve Hand Gear.
Telemotor Line to Bridge.
Deep Load Line
Ballast Draft with 50% Fuel Capacity Aboard.

COMPARATIVE DIAGRAMS SHOWING 4500-TON DEADWEIGHT MOTORSHIP FITTED WITH A 3-PLANE FLETTNER RUDDER AND ONE OF ORDINARY TYPE.

and the center plane supports the two wing planes by means of strong transverse members which maintain all three planes rigidly parallel under all conditions.

Among the first to appreciate the practical advantages of the 3-plane Flettner rudder was the Germania Werft of Kiel, and this firm were also quick to realize that unless the Flettner rudder were incorporated in a vessel whose hull had been specially designed from the first for such a rudder, the best results might, perhaps, not be achieved. Accordingly the Germania Werft took the first available opportunity of embodying the 3-plane rudder in new constructions and this opportunity was presented when the Reederei H. C. Horn, Flensburg, some time ago placed an order for a 4500 tons dead weight motorship. With the owners' approval the 3-plane rudder was accepted in the original specifications drawn up and mutually agreed. The new motorship, the *Therese Horn*, which was recently completed at Kiel,



PROPOSED ARRANGEMENT OF 3-PLANE FLETTNER RUDDER FOR A TWIN-SCREW VESSEL.

proceeded on sea trials on September 5. These trials were noteworthy inasmuch as they represent the first Flettner rudder installation for which the hull has been specially designed and constructed, as it was also the first 3-plane rudder to be installed on a seagoing merchant vessel.

The trial trip was attended by a large number of technical experts drawn from the leading shipbuilding and marine engineering establishments in Germany and abroad and also by numerous owners, superintendents, consulting

engineers and classification surveyors. The trials are stated to have fulfilled the highest expectations of builders and designers and to have confirmed in a remarkable degree the earlier model experimental data derived from tank performances. During the trials the inventor delivered an introductory address and subsequently the guests were able to appreciate for themselves to what extent the claims of the inventor were borne out in practice. With the vessel proceeding on a straight course at maximum speed ahead the rudder was put hard over and the vessel is reported to have performed a complete circle of a diameter equivalent to less than two lengths. This evolution was repeated several times, port and starboard, with uniformly satisfactory results and it has been stated that when traversing this circle it was possible to bring the vessel back to a dead straight course within 15 to 20 seconds. The advantages of sensitive maneuvering qualities such as these in narrow congested waters or harbors will be at once manifest.

Not the least important feature of the Flettner rudder is the complete elimination of heavy and expensive mechanical steering gear. This feature has a dual advantage as in the first place the initial cost of the vessel is less and secondly a certain reduction in fuel consumption and periodical steering gear repairs will be realized. These are features which are likely to appeal to the harassed shipowner of today. It has been argued by opponents of the Flettner rudder that in virtue of its shape and inasmuch as two of its planes are off the center line it may be vulnerable to damage in a seaway or in the case of grounding. The former objections have yet to be proved in practice and with regard to the latter it may be observed that the lowest point of the 3-plane rudder surface is about 3 feet above the keel line so that grounding should not normally cause the damage stated. A further advantage resulting from the use of 3-planes was demonstrated in the recent trial trip. With the vessel going dead slow and with practically no way on, the steering qualities were excellent. Furthermore, with the vessel completely stopped, only a few revolutions of the propeller were necessary to swing the vessel's head round when helm was given.—"Shipbuilding and Shipping Record," Sept. 25, 1924.

THE VACUUM—THERE'S SOMETHING IN IT.

By Dr. W. R. Whitney,

Director, Research Laboratory, General Electric Company.

We humans want better minds, broader horizons, and greater understanding. Scientists everywhere are at work in their respective fields searching for new truths to improve the process by which our minds, our horizons, our powers, and our outlooks grow.

When, in the middle ages, the great cathedrals of the world were being built, the mentality of men seems to have been directed to systematic subordination of creation rather than to active appreciation of it; to acquiring salvation, or "safety first," rather than knowledge or understanding. Today scientists, in seeking truth, think of "safety first" last.

We are told that if the total age of mankind be expressed as the life of a man of fifty years, and if he then looks back upon his progress, he will see that what marks his greatest advancement are events occurring in the most recent years. For example, such a fifty-year man now sees that he had not learned to scratch the simplest records on stone until his forty-ninth year. All the immense advantages of printing have existed only three months for him. He has only just learned how to pass along what he has learned. The uses of steam, which now seem so necessary, were acquired only three or four days ago. The uses of electricity (street cars, lamps, and telephones), which did not actually begin until about 1880, arrived the

day before yesterday to this fifty-year man. The automobile, radio, X-rays, radium, and most of the things which occupy our interests today, were actually discovered on this particular fiftieth birthday.

The power outside of his own muscles, which during the past two or three months he has learned to control, has grown to nearly twenty horsepower, or two hundred man power, for every man in the country. Therefore it makes only one-half a per cent difference whether all the men work like horses or not. But *guidance* of power is man work, because there are no machine mentalities. Almost everything but thinking may be artificially done, but knowledge and understanding must be actively sought and used. Man is the only animal that can do this.

Scientists know that research merely discloses new parts of the infinite unknown. Paradoxically, the enticing, helpful "unknown" increases as men continue to subtract *from* it. Progress in every line of experimental science follows the same law. The apparently narrow path gradually expands into unlimited, unexplored territory. With his new tools and his increased speed of communication, man finds that he can advance into the unknown faster than his ancestors could, and children seem to learn more rapidly than he did when he was young. The scientists of the twentieth century are legion. But scientists were anathema a short time ago. There is today more chemistry in the atom than there was in all "inorganic" chemistry a few years ago. There is more in the "sugars" now than there was in all "organic" when the writer studied it. There is more immunity in blood, and more heredity in the microscopic chromosome, than there was in all biology until recently. There is more crystal structure research by X-rays now than research in all mineralogy when Agassiz came to America.

In the preparation of an article the purpose of which is to direct attention to the interest connected with research work in general, there is special reason for selecting as the subject such a narrow field as the vacuum. The writer wishes to show that in a vacuum, of which one might say "There is nothing in it" (and surely less than in anything else), there is, indeed, an endless amount of interest and utility. The American public now buys over a million dollars worth of glass vacua a week, but that is the least interesting part of the subject.

Everybody pretends to know that "Nature abhors a vacuum." But he who started that tale merely meant that a good vacuum was hard to produce. As probably no one has ever made a vacuum with less gas-molecules in a cubic inch than there are people in the world, we can maintain that perfection in vacua is still precluded by nature.

Some studies in vacua will now be reviewed so as to create the impression that any one may well do research work not only there but also certainly anywhere else with pleasure and profit.

We all know that we see at night largely by the aid of vacuum lamps. Through other vacuum lamps, called X-ray tubes, we also see through opaque bodies. The light which illumines our microscope specimen has its analogue in the X-ray light which shows us the crystal structure of matter and the electrical formulas of chemical atoms. Our trans-continental wired telephony is possible through vacuum tubes which, in various forms, also permit our radio broadcasting and radio reception from the most remote stations. The workman who keeps his drink hot or cold in a thermos bottle is clearly indebted to Sir James Dewar's application of the vacuum but the scientist is still more indebted to it. All our steam power plants, including turbines, owe their success to vacua.

The latest arrival in the family of the chemical elements (hafnium) was discovered by trying it in vacuum as an X-ray target. Our list of possible chemical elements was rounded off by Moseley's study of the X-ray spectra just in time to meet the wonderful discoveries of J. J. Thomson and Aston, all made in vacua. The latter showed that most of the elements, supposedly

simple, are still mixtures of two or more similar elements (isotopes), while Professor Thomson's experiments disclosed by the positive ray method a whole series of new atomic compounds.

While the vacuum was not essential to the work of Millikan in isolating the electron, yet the earlier work by Thomson and others, and much of the recent work on this ultimate constituent of matter, has been necessarily carried out in vacuum. Today there seems to be no end to the studies which can be based on the fact that an atom or molecule of material may be separated electrically into a positively charged ion (carrying most of the mass) and a negatively charged electron (carrying most of the current).

ELECTRICITY.

Without sacrificing historical truth too much, it may be asserted that electricity at rest was the first kind known. It then seems logical that electricity in simple direct motion should later appear and that still later we should find various directions and rates of its motion, if it moves at all. It will interrupt this line of argument if we doubt the existence of electricity as a thing, or question the existence of such different kinds of it as static and dynamic. The electricity of rubbed amber was the first and stationary kind (if it is admitted that prior to Thales such things as lightning were something else). When electricity first moved through metals the process was looked upon as a simple directed flow which proceeded until the charged body had delivered its charge.

CURRENTS.

This direct flow of electricity (a current) was strengthened in its hold on our conception by the great number of different chemical current-producers which followed the controversies between Galvani and Volta a century and a quarter ago. Primary and secondary or storage batteries without number were soon discovered. The current from such batteries was exactly like that which the magneto-electric machines produced, when these were developed, after Faraday had shown the effect of moving a wire through a magnetic field.

The direct current of such generating machines was much later followed by the alternated flow, and soon alternating-current generators were made for different rates of reversal of direction. Sixty-cycle and twenty-five-cycle currents are now common and the user takes his choice. These frequencies were once accidents of convenience and economy. For some uses other widely different frequencies of alternation are very desirable, as in radio, where 1,000,000 cycles are common.

Now the phenomena which have been found in vacuum tubes promise to give complete control over all these details of kind and frequency of current.

ELECTRICAL CONTROL.

As will be more fully shown later, when a unidirectional current meets vacuum tubes as though it would pass through them, it must find one particular kind of a tube, and the current's direction must be right, because some tubes will let it through only when it is both unidirectional and in the proper direction. These in themselves are rather remarkable things to expect of a vacuum, but as usual the truth exceeds the expectation.

CATALYSIS.

To give some idea of the extent to which vacuum studies may affect remote fields, mention may be made of chemical catalysis, the secret of most reactions of life. For example, it is known that mercury vapor in a vacuum, when illumined by light of a certain wavelength, will absorb that

light and turn the energy over to hydrogen, if hydrogen be present, so that this, in turn, will chemically reduce such substances as cold copper oxide. Here is a new kind of chemical process. It is the kind we have needed in order to begin to explain some of those life-reactions which vegetation discloses. That is, similar facts will probably be found to contain the explanation of the catalytic action of sunlight on growing plants.

And so the studies of phenomena in vacua may lead us into the most widely separated fields. The experiments described in the latter part of this article are illustrations of this fact, and are thus not chosen to be very closely related.

KINDS OF VACUA.

One might say, as in our school-day essays, "There are different kinds of vacuum too numerous to mention" and then proceed to mention them all. However, only a few cases will be selected for the purpose of illustration. A certain kind of vacuum is good enough for incandescent lamps because other factors besides the quality of its vacuum determine the death, or limit the performance, of a lamp. But while formerly the incandescent lamp represented the very highest skill then reached in vacuum production, there are now other commercial vacua which are necessarily quite superior. This is true of good thermos bottles, X-ray tubes, and radio tubes.

We should first, therefore, give brief consideration to the incandescent lamp, and note a few characteristics. Its low vacuum early disclosed electric cross currents which would not have been found in much higher vacua. Study of these currents has led not only to the vacuum tubes used in radio, but also to such remote disclosures as that distillation or evaporation of solids in vacuum proceeds in straight lines. The electrical currents in poor vacua are best known in luminous Geissler tubes, Moore tubes, and Claude tubes, as seen on the streets of Paris, for example. They are themselves a large subject, but consideration will be given (later on) only to one historically important experiment with them, the Hittorf experiment.

HIGH VACUA.

Electric currents through vacua, where gas is so completely removed that it has no appreciable action, are more strictly a part of this article. After the work by which the individual and indivisible negative electrical charge or electron was defined, it seemed quite fitting, though unexpected, to learn that these negative charges were exuded by hot bodies. The result of this disclosure formed the basis for most of the modern electrical phenomena in vacua. The activities of electrons are apparently the cause of most electrical and chemical processes. Their motion constitutes electric currents, and the currents are determined and controlled by voltage or potential difference.

Bees might illustrate electronics, though it's admittedly bad policy to push bees too hard. If bees represent electrons, then matter in general becomes the hives. When the hives are cold or in the dark the bees stay inside. Under the effect of heat or light the bees are induced to come out. Similarly highly heated matter, such as tungsten, exudes electrons; and at ordinary temperatures light induces electrons out of metallic potassium, for example.

What are the bees or electrons going to do after they are out of the hive (or the metal)? That depends on impulse or pressure. They will fly in that special direction which tends to relieve that particular pressure. They proceed down the gradient. The electrons coming out of the metal (because of heat or light) will also fly in that special direction which will tend to relieve the particular electrical pressure. They also proceed down the gradient. But if there is no particular impulse or gradient both bees and electrons hang around the source.

A lot of electrons, like bees, flying in a common direction, becomes a current or cathode ray. We might let them fly from the hot or lighted spot into a cold or dark one, and until we heated or lighted the latter they would not come out. This unidirectional current corresponds to what takes place in rectifiers and kenotrons where we have impulses in two opposite directions between one hot and one cold electrode in vacuum. If the hives (or the electrodes) are equal in temperature or illumination, no differential in current is possible, and with alternating impulses alternating instead of pulsating direct current passes. Thus bees illustrate two-electrode vacuum devices.

In the three-electrode vacuum tubes the third electrode is a sort of grid, or open fence, located between the hot and cold electrodes. This grid lets electrons pass freely except when it is negatively charged. Thus also the bees would pass through a wire fence when they might be stopped if their impulse to proceed could be suddenly removed there. The negative charge is used in the three-element tube to alter the intention of the migrating bees or electrons as often as desired, and this with the rapidity of light.

In the case of radio, it is the impulse from the antenna or loop, changing with every delicate change of voice-current or code-current, which, when led to the grid, charges it and thus controls the currents within the receiving tube. These controlled currents do the work in the telephone.

Everybody knows that radio tubes are very sensitive. One cat-power of electricity used in New York actually puts the impulses into a receiving outfit in San Francisco, and at the same time it also puts the identical impulses into millions of other receivers. But some appreciable energy must be used by each receiver to direct the local battery which operates the head sets. This minute quantity of energy may be made significant as follows:

If a house-fly climbs up a window pane one inch, he does a definite amount of work in lifting his body that much. If this work constituted the supply fed into the receiving tube from space, it would suffice to actuate the outfit continuously for a quarter of a century. This might interest a future student of telepathy, if the time comes to determine how far the energy of one thought may influence thought in a distant brain. It has not been possible thus far to determine the quantity of energy which is expended in thought. Just keeping alive transfers so much energy into heat that the additional energy transferred when we think has been too small to detect. But it need not be small compared to the power sensitivity of a radio set.

VACUUM TUBES.

The mere names of modern vacuum-tube applications of electrons are legion. The kenotron is a vacuum tube which changes high-voltage alternating current into direct current, and it has its counterpart for low voltages in the battery-charging tungar rectifier and the mercury rectifier. The various radiotrons, receivers, and amplifiers of radio are also the most direct applications of the action of negative electrons in good vacua. But X-ray tubes must also be considered in this connection, because X-rays are the result of the "bump," if you will, of rapidly moving electrons in vacua against the atoms of matter.

Electrons in motion are also directed and controlled by electromagnetic, as well as by electrostatic, fields. Therefore the magnetron and axiotron have to be included. Because the mere illumination of such metals as potassium (like the high heating of other metals, such as platinum and tungsten) causes them to emit electrons, the photo-electric cell has to be included in our illustrations.

The electrons within the vacuum, as in a radio detector tube, obey the inconceivably feeble electrical impulses received by the antenna. Conversely,

the motion of electrons sets up impulses and waves in space. In other words, the vacuum detector may be made a radio wave generator. Such tubes are used in broadcasting stations.

ETHER WAVES.

The wavelengths of the magnetic waves produced by the changes of motion of the electrons are very long in the case of radio (say 100 to 10,000 feet), millions of times shorter in ordinary light, and millions of times shorter still in X-rays, but they are all in the same medium and all due to motions of electrons.

SPECIAL EMISSIONS.

Finally, to give an impression of the distance such work has gone, there should also be added the case of a one-atom-deep layer of thorium on tungsten in a vacuum and its effect on electron emission.

EXPERIMENTS.

LAMPS.

The first great use of vacuum was in incandescent lamps. If such a lamp is burned at much higher than its rated voltage, it lasts for a few minutes only, but it gives a light perhaps five times as efficient as we usually see. The lamp dies because the tungsten vaporizes or melts. The vacuum is not at fault. It is because of these limitations that commercial lamps are so made that they burn at normal voltages on proper circuits an average of one thousand hours. If we were satisfied with a shorter life we could have a more efficient lamp, but experience has shown that we would be unwilling to use the short-life lamps in order to secure the added efficiency. To increase the efficiency without shortening the life of lamps much study has been made of distillation of metals in vacua, and of methods for returning the distilled metal to the filament. Much has also been done to make the deposit on the glass invisible or white, so as not to interfere with light transmission. Naturally we are always on the lookout for metals, such as the newly discovered hafnium, which might possibly live longer as a filament than tungsten now lives. Thirty or forty years of research work had been spent on high-vacuum incandescent lamps before Dr. Langmuir showed us how to make still better lamps by putting back into the vacuum gases like argon and nitrogen.

DISTILLATION.

When the material of a filament distills in vacuum it does not meet interference to the motion of its molecules, and the distilling substance proceeds in straight lines from the heated source. This is often observed when an incandescent vacuum lamp arcs or burns out. Metal shadows of interior parts of the lamp are then often cast onto the walls. This is shown more clearly when a metal like gold is evaporated from the surface of a tungsten filament in vacuum. By the interposing of a design, as for instance, the star in the left-hand bulb of Fig. 1, a shadow in gold is cast on the glass as shown on the right-hand bulb of Fig. 1. This simple phenomenon is mentioned because it fits in with the kinetic theory of gases and explains many things observed in vacua. The "mean free path" of molecules or atoms is very long in good vacua, and so straight line distillation occurs.

HITTORF BULBS.

In the historic Hittorf experiment two vacuum bulbs, each carrying an electrode, were joined together by two glass tubes, one very short and one exceedingly long, as shown in Fig. 2. When electric current was passed from one bulb to the other, it evidently chose the longer instead of the shorter path, because the longer tube became highly luminous, while the shorter one did not. This is a quality of electrical conduction in vacua where small quantities of gases still remain, but it cannot be gone into here.

EDISON EFFECT.

Another experiment shows the historic Edison effect and its relationship to pure thermionic currents. Where it was once thought that nearly visible particles of the filament were shot across the space between filament legs in vacua, now we recognize, in very high vacua, only the unidirectional motion of negative electrons.

RECTIFIERS.

In vacuum tubes like the kenotron these electrons pass from the hot filament to an electrode commonly called the "plate." This pure emission current is the basis for the so-called rectifiers because only when the filament is negative does any current flow across the space. When gases are present greater currents may be carried, because by the ionization of the gases the moving electrons produce new conductors from the gas molecules. Thus the tungar rectifiers, containing a little argon, and the older mercury vapor rectifiers involve the same principle. Without some gas present the negative electrons, by their very concentration, constitute a space charge which limits the current. This space charge is removed by the ions produced within the gas when present.

RADIOTRONS.

When we interpose a grid or wire screen between the hot filament and the plate of the two-electrode tube or rectifier, we have what we now so commonly use in radio for receiving, for amplifying, and for production of high-frequency currents. The discovery of the controlling or triggering action of the third or intermediate electrode was made by De Forrest. A negative charge applied to this electrode or grid interrupts or modifies the electron stream, the current, from the hot filament to the plate. As it takes almost no energy to charge this grid (little more than a "token" of energy, or voltage) the slight power from a radio antenna in its fluctuation may be used to control or to trigger or to let through corresponding jolts of greater energy, which are in turn supplied by some local battery. In the experiment illustrated in Fig. 3, an ordinary incandescent lamp is lighted by current which is passing through a three-electrode tube from hot filament to plate. Its light indicates this current. The grid, or antenna wire, is sticking up from the tube so that it can "pick up" electric charges from space. A small negative charge, produced on a rod of insulating material by merely rubbing it with a piece of paper, causes the lamp to go out or to light up as the charge rod is brought near to or removed from the exposed end of the grid wire.

MAGNETRON.

As the negatively charged grid cuts off the current of the three-element tube, so an external magnetic field will also do it in the two-element tube, by so influencing or directing the moving electrons that they cannot reach

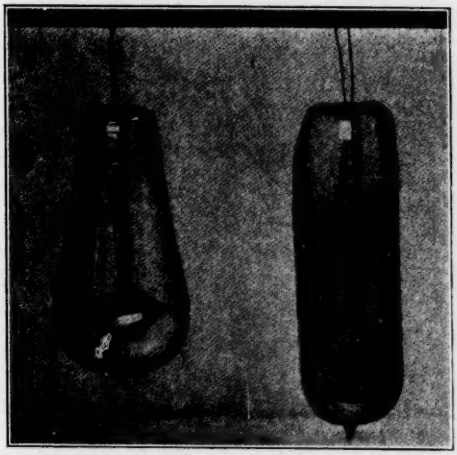


FIG. 1.—A DEMONSTRATION OF THE STRAIGHT LINE DISTILLATION WHICH TAKES PLACE IN GOOD VACUA. IN THE RIGHT-HAND BULB, WHICH HAS INSIDE ELEMENTS DUPLICATING THOSE IN THE LEFT-HAND BULB, THE SHADOW PATTERN OF THE INTERPOSING STAR SHOWS CLEARLY IN THE FILM OF METAL DISTILLED ONTO THE INTERIOR WALL.

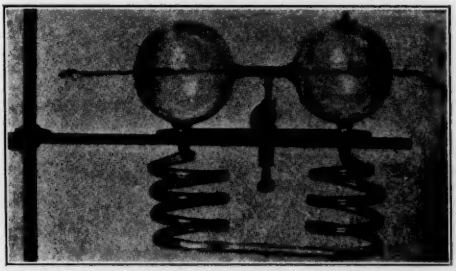


FIG. 2.—HITTORF EXPERIMENT IN WHICH AN ELECTRIC CURRENT PASSED FROM ONE BULB TO THE OTHER AT LOW GAS PRESSURE PREFERS THE LONGER OF THE TWO TUBE PATHS.

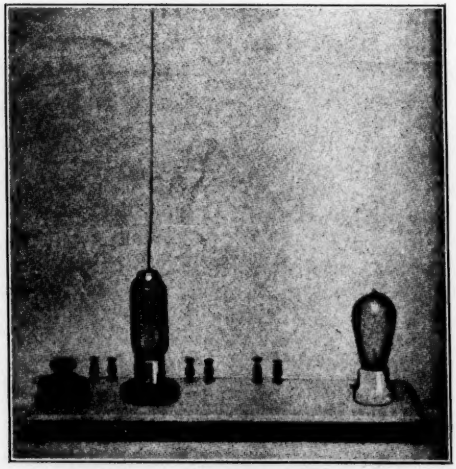


FIG. 3.—A DEMONSTRATION OF THE RADIOTRON PRINCIPLE. ELECTRON CURRENT FROM THE HOT FILAMENT TO THE PLATE OF THE RADIOTRON TUBE ON THE LEFT IS MADE TO LIGHT THE INCANDESCENT LAMP AT THE RIGHT. THIS CURRENT CAN BE STOPPED BY INDUCING A NEGATIVE CHARGE ON THE VERTICAL ANTENNA WIRE WHICH IS CONNECTED TO THE GRID.

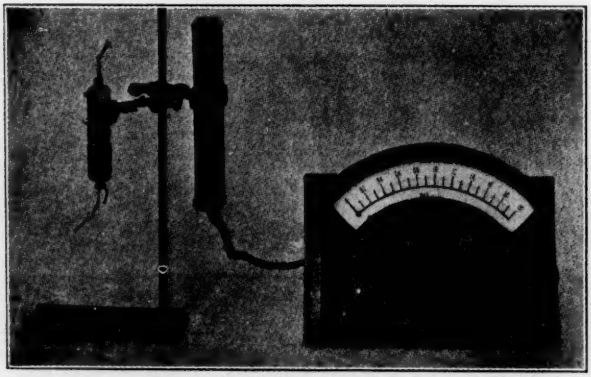


FIG. 4.—A DEMONSTRATION OF THE SENSITIVENESS OF A MAGNETRON TUBE TO A MAGNETIC FIELD. WHEN THIS TWO-ELEMENT TUBE IS SUITABLY ARRANGED, ITS POSITION WITH RESPECT TO TERRESTRIAL MAGNETISM INFLUENCES THE METER READING.

the plate. Such a device is called a magnetron. The particular magnetron used in this experiment, Fig. 4, consisting of a cylindrical plate in a vacuum and a central filament within it, has a coil of wire wound on the outside of the glass. A feeble current sent through the coil from a battery may be made to set up within this coil a magnetic field about equal to that of the earth's magnetic field, so that by moving the whole apparatus about in space (thereby at one time adding to the earth's field, at others opposing it), the magnitude and direction of the earth's magnetic field can be disclosed by a meter which indicates the resultant current. When the magnetron points towards the north pole, the meter shows no current, while in other positions currents are measured.

AXIOTRON.

Another useful vacuum tube of this type is one in which the magnetic field of the filament current itself becomes great enough during each current cycle to deflect the electrons so that they will not reach the plate. By this tube, "the axiotron," the frequency of an alternating current may be doubled or direct current be changed into alternating.

PHOTO-ELECTRIC CELL.

Another vacuum tube is the photo-electric cell. As shown in Fig. 5, one of these may be connected with a relay to a lamp so that when light shines upon the cell, the burning electric lamp is extinguished, and relighted on cutting off the light from the photo-electric cell. In other words, it turns on the lamp when it is dark, and turns it off when it is light. This depends on the fact that some metals, like potassium, emit electrons when light falls upon them. These electrons in vacuum constitute a current when they are made to move by the electrical impulse. To repeat; applying potential to a vacuum tube having a potassium electrode and another electrode for leading off the current, causes current to flow in the form of negative electrons from the illuminated metal, and this current actuates the electrical switch which turns off the lighting current of the burning lamp.

THE NERVE.

This leads to the next, an ambitious physiological experiment. As illustrated in Fig. 6, a photo-electric cell may be made to represent a crude "eye" which is connected to a so-called "nerve" leading to a "brain." The nerve is merely a long box having electrical capacities and suitable amplifiers or three-electrode tubes within. The capacities serve to slow down the apparent rate of flow of the feeble current from the artificial eye so that indicating lamps along the top of the box or "nerve" light up, one after another, as the impulse from the "eye" passes along that path, or the nerve. After the last lamp is thus lighted by these amplified currents an electric bell rings to indicate reception at the "brain" end of the circuit. This is not offered as a reliable replica of the real nervous system, but as an application of the vacuum tube which amplifies the slight energy available and necessary for the experiment. With such slight energy it becomes practicable to show the delayed transmission and reception which is necessary for an illustration of nerve action. Nerve impulses travel much slower than electricity usually does, and this low speed was one objection to visualizing nerves as electrical conductors until Crehore and Williams showed that nerves might be naturally so constructed as to transmit slowly.

X-RAY.

In order to show another vacuum product (the X-ray tube) an effect of X-rays may be demonstrated which is not usually thought of in connection with X-rays. Three rubber balloons, suspended close together by long cords, are first charged electrically by friction. They then repel each other and stand stationary in space as at the corners of a large triangle, Fig. 7. As soon, however, as a feeble beam of X-rays is projected towards them in the manner shown in Fig. 8 they quickly discharge and fall to their original position in contact with one another. X-rays ionize air or make it conducting so that the balloons cannot retain their mutually repelling charges. This is the basis of a method for measuring the intensity of beams of X-rays.

SPECTRA.

Another application of X-rays is in the study of internal structure of crystalline chemical compounds and elements. Cathode rays, currents of negative ions in vacuum, when speeded up by high voltage, produce by their impact X-rays which are characteristic of the material on which they impinge; one may say characteristic of the mass of the atom of the substance of the target. It is through this fact that the X-ray spectra of the elements considered as to wavelengths are arranged in the same order as the atomic masses in the periodic table of Mendelejeff, and by this very method the newest known element, hafnium, has been recently added to the known metals. Referring again to our bee analogy, let a fast flying swarm strike bells so hard that they make them ring. From the sound or musical notes we guess roughly the sizes of the bells. We could thus place them in their musical series. The sound corresponds to the X-rays produced when the bees are electrons of cathode rays and sound waves are ether waves. The mass of the bell is disclosed by the tone or frequency; the mass of the atom, by the same sign, is disclosed by the ether wave-frequency. When a certain mineral was used as a surface for the electrons to hit, a new musical note in the ether was found. It was recorded photographically. Its place in the scale of elements had been predicted as accurately as middle C on the piano might have been predicted if it had never been heard.

With higher voltages the velocity of the cathode rays (or electrons) always increases. In the X-rays thus far produced, however, the penetration or transparency is practically limited to about a quarter of an inch of lead. It is interesting to note that the similar rays from radium, the so-called gamma rays, can penetrate nearly a foot of lead. This corresponds to an exceedingly high electromotive force. Thus radium rays (gamma rays) might be made in vacuum X-ray tubes if millions of volts were applied.

ATOM LAYERS.

The apparatus shown in Fig. 9 (devised by Dr. Hull) is essentially a two-electrode vacuum tube, the tungsten filament having a little thorium in it. At a certain very high temperature this thorium rapidly diffuses to the filament surface. This thorium surface then has the peculiar power of emitting electrons a hundred-thousand times as rapidly as pure tungsten at the same temperature. An ordinary lamp may be lighted in this experiment by letting this thorium electron-emission current flow through the lamp filament. The vacuum tube containing the thorium-coated tungsten also contains a little gas. When the lighted lamp is short circuited for an instant by a switch, the potential on the vacuum tube is thereby greatly increased, and this causes positive ion bombardment of the filament and thus tears the thorium all off the tungsten surface, so that very few electrons are being emitted, that is, those characteristic of pure tungsten at that temperature.

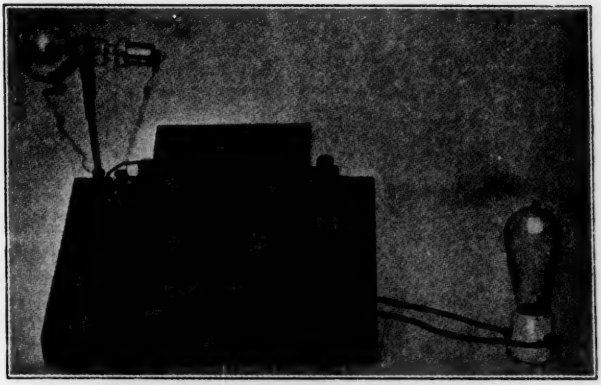


FIG. 5.—A DEMONSTRATION OF THE ACTION OF A PHOTO-ELECTRIC CELL CONNECTED THROUGH A RELAY TO AN INCANDESCENT LAMP. WHEN NO LIGHT FALLS ON THE CELL, THE LAMP BURNS; WHEN THE CELL IS EXPOSED TO LIGHT, THE LAMP IS EXTINGUISHED.

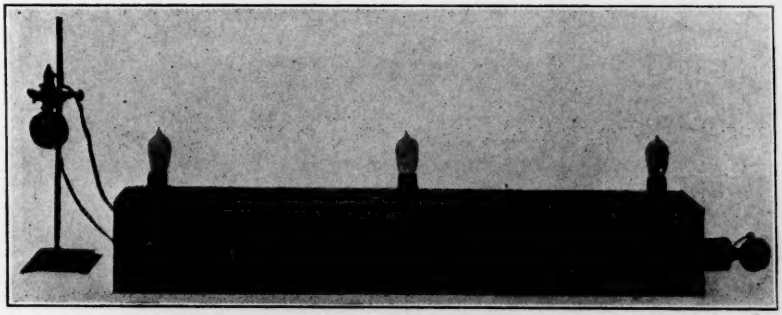


FIG. 6.—APPARATUS CONSISTING OF A PHOTO-ELECTRIC CELL, INDICATING LAMPS, CAPACITIES AND AMPLIFIERS (IN BOX), AND A DOOR BELL ARRANGED TO DEMONSTRATE THE RETARDED TRANSMISSION OF AN IMPULSE, SUCH AS NERVE ACTION.

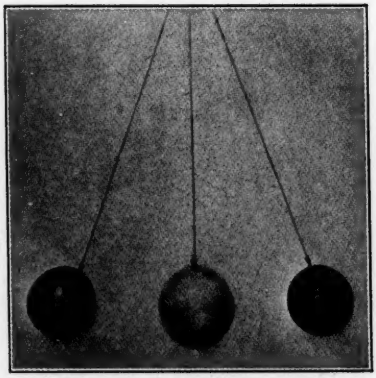


FIG. 7.—THREE SUSPENDED RUBBER BALLOONS REPELLING EACH OTHER TO A TRIANGULAR FORMATION AS THE RESULT OF BEING ELECTROSTATICALLY CHARGED. SEE FIG 8.

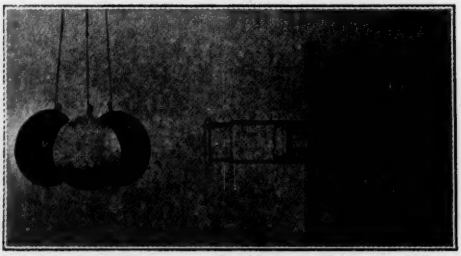


FIG. 8.—THE BALLOONS ILLUSTRATED IN FIG. 7 ARE HERE SHOWN IN THEIR NORMAL POSITIONS AS THE RESULT OF THEIR CHARGES HAVING BEEN DISSIPATED BY A FEEBLE BEAM OF X-RAYS.

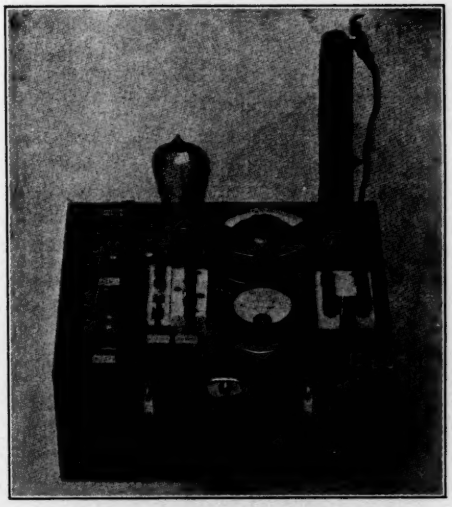


FIG. 9.—APPARATUS DEVISED TO DEMONSTRATE THE CHARACTERISTICS OF ELECTRON EMISSION FROM A THORIATED TUNGSTEN FILAMENT.

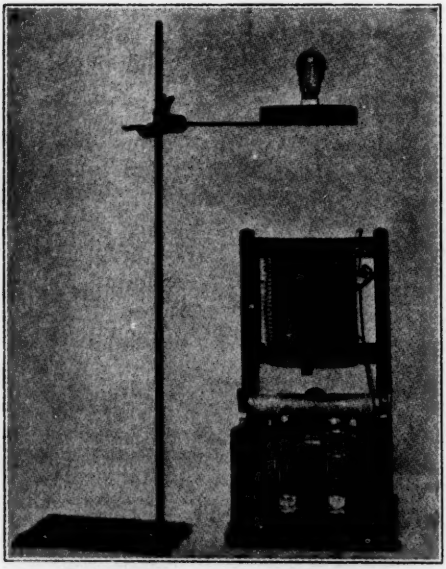


FIG. 10.—A DEMONSTRATION OF THE TRANSMISSION OF ENERGY THROUGH SPACE BY MEANS OF WAVES OF SEVERAL MILLION CYCLES. THE LIGHTED LAMP ON THE STAND RECEIVES ITS ENERGY FROM THE SOURCE LOCATED BENEATH.

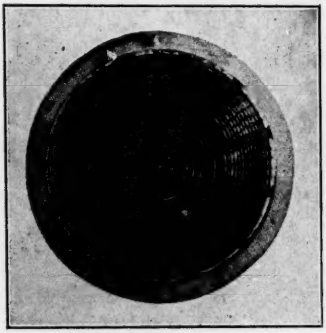


FIG. 11.—THE HEWLETT LOUD SPEAKER. NO HORN IS NEEDED ON ACCOUNT OF THE LARGE SIZE OF THE DIAPHRAGM, 26-INCH DIAMETER. THE ABSENCE OF THE HORN AND THE DISTRIBUTED EXCITING COIL MAKES REPRODUCTION POSSIBLE WITH A HIGH DEGREE OF FAITHFULNESS. FOR ITS OPERATION, A SET OF VACUUM TUBES AMPLIFIES THE MICROPHONE CURRENT.

Then the load (the lamp) can no longer be carried by the electron current. To repeat, the heavy positively charged gas ions under the impulse of the raised electrical potential act like a powerful sand blast and effectively clean the thorium from the tungsten, thus greatly reducing the emission current. By highly reheating the filament for a few seconds only, a fresh layer of thorium may be diffused to the surface of the filament from within the tungsten, so that then at the previous lower temperature the load or lighting current for the lamp is carried as before by electrons emitted from the thorium surface. This is a proof of the production of a layer of thorium on the tungsten only one atom deep. The electron current from thorium on tungsten is greater than from pure tungsten, and also than pure or massive thorium, and is maximum when the single atom layer is present. This is confirmed by experiments on partial recovery of the surface and supported by thousands of successive repetitions of this experiment on one filament.

HIGH FREQUENCY.

By means of an apparatus such as shown in Fig. 10, an ordinary incandescent lamp may be lighted by being brought within a foot of a coil which is carrying a current of several million cycles. This, produced by pulsations, is about as near to wireless transmission of power as anything we now have. This high-frequency principle is also being used by Professor Northrup of Princeton for special electric furnaces. In these, the induced currents in the material of the crucibles or the material to be heated generate high temperature through local resistance.

LOUD SPEAKER.

Fig. 11 shows a loud speaker of unusual design. It consists of a 26-inch flat conducting disc, in a magnetic field, the vibrations of which correspond to the voice currents and reproduce the sound waves without the intervention of a horn. To operate this device, vacuum tubes are made use of as follows:

Sound waves entering a microphone cause feeble electromotive forces to be generated in the microphone. These feeble electromotive forces are applied to the grid of a pliotron and cause relatively large variations in the electric current flowing between filament and plate, which in turn are used to secure larger electromotive forces to be applied to the grid of another pliotron. By the use of several amplifying pliotrons the original feeble electric currents are multiplied several thousand times and supplied to the loud speaker which reproduces the original sounds with many times the original volume and great faithfulness of quality. To operate this pliotron amplifier requires a direct current of several hundred volts. This may be obtained by first transforming power from the ordinary alternating-current lighting circuit to a relatively high voltage, next rectifying this high-voltage alternating current by means of kenotrons, and finally smoothing out this pulsating current by means of appropriate electric circuits.

The high degree of faithfulness of reproduction realized in this loud speaker is due partly to the absence of a horn, eliminating horn resonance (one of the usual sources of distortion in a speech reproducer), and partly to the method of vibrating the diaphragm by forces which are distributed fairly uniformly over its surface, instead of acting upon it in a very limited region, as is the case in most other loud speakers. This feature eliminates rattling and ringing of the diaphragm or the production of high overtones by the diaphragm vibrating in its partial nodes.—"General Electric Review," July, 1924.

FIRST PAN-AMERICAN STANDARDIZATION CONFERENCE HERALDS GREAT ADVANCE.

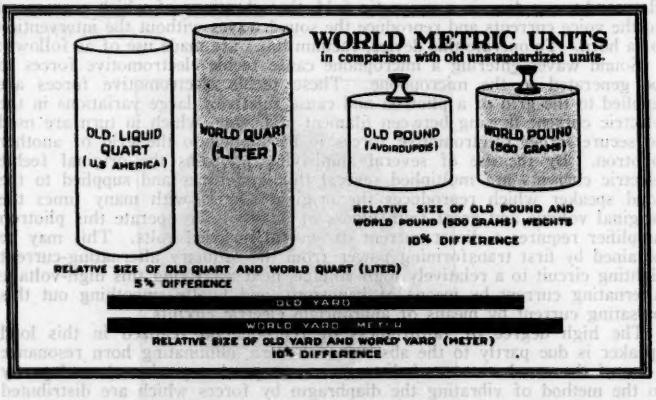
COMMODITY QUANTITY STANDARDS TO BE UNIFIED THROUGHOUT ALL THE AMERICAS.

An epoch-marking event is the First Pan-American Standardization Conference to be held at Lima, Peru, in December, 1924. This conference, called with the co-operation of all the American republics, will determine upon uniform specifications and quantity nomenclature for raw materials, merchandise and commercial equipment. The sessions also will take up the general subject of standardization—its principles and its importance in the economic development of the world, with particular reference to inter-American trade. Accomplishments of European countries in establishing uniform standards will be reviewed.

As nearly all the participating countries produce raw materials in excess of finished materials, special emphasis is to be placed on trade standards for raw and crude products. National and international specifications will be considered for such commodities as grain, cotton, cocoa, coffee, rubber, sugar, tobacco, vegetable oils, lumber, hides and skins, wool, meat and dairy products. Industrial standardization and specifications for finished materials likewise will be studied. The creation of uniform inter-American and international standards is contemplated.

A most important section of the conference will deal with the standardization of containers, packages, and packing. The need for uniform containers will be stressed. A thorough study of this simplification has been urged in advance, so that delegates may unite on uniform sizes of containers.

Not only quantity standards will be established, but also standards of quality—these being, in truth, inter-dependent: for grades and classifications must be based on a uniform measure.


WORLD METRIC UNITS
in comparison with old unstandardized units.
OLD LIQUID QUART (U.S. AMERICA)
WORLD QUART (LITER)
OLD POUND (AVOIRDUPOIS)
WORLD POUND (500 GRAMS)
RELATIVE SIZE OF OLD POUND AND WORLD POUND (500 GRAMS) WEIGHTS
10% DIFFERENCE
RELATIVE SIZE OF OLD QUART AND WORLD QUART (LITER)
5% DIFFERENCE
OLD YARD
WORLD YARD METER
RELATIVE SIZE OF OLD YARD AND WORLD YARD (METER)
10% DIFFERENCE

The First Pan-American Standardization Conference has vital significance in relation to the world-wide movement for metric standardization of weights and measures. All American republics are already on the metric basis with the exception of the United States of America. Metric advocates declare that the forthcoming event offers an unprecedented opportunity for the United States of America to advance logically to the metric commodity

units likewise. Accordingly, it is being urged that the topic of metric standardization hold prominent place on the program at the Lima conference and that a resolution be passed calling upon the United States of America to standardize with all the rest of the American republics upon the world units.

Throughout the United States of America progressive elements are organizing to direct the attention of the delegates to the urgency of the metric issue, and to awaken the American public to further participation in the metric advance.

It is confidently expected that as a result of the action of the First Pan-American Standardization Conference, meeting in December, the Congress of the United States of America (which also convenes in December) will be moved to enact definite legislation providing for a gradual transition to the metric commodity quantity standards—world yard (meter), world quart (liter), and world pound (500 grams). When this transition shall have been brought about, all the American republics will then be on a uniform basis for interchange of commodities.

The resultant savings in elimination of labor, time-waste and confusion will aggregate billions of dollars, which can be employed usefully in the upbuilding of civilization.

SUBMARINE CONSTRUCTION.

As a result of the Washington Agreement for the limitation of naval armaments, the construction of submarine craft, was left unfettered, and any country was free to build this class of vessel to any size and in whatever quantity the various Powers desired.

Accordingly we hear now and then of great submarines being launched, just as in the old days we heard of the launch of battleships larger than any that had gone before. The first of a new class of American submarine, of which nine will be built, has just taken the water. It is 340 feet long—twice as large as any previously in the United States Navy—has a surface speed of 21 knots, a surface displacement of 2164 tons, a complement of 87, and is believed to be capable of twice crossing the Atlantic without replenishment of fuel or supplies.

Whatever may be said, then, on the much-debated question of the battleship *versus* submarine, clearly both the United States and Great Britain—for we, too, have not been backward in this matter—believe that the submarine has an important future, and that it is worthy of some of the attention that naval designers used to lavish so affectionately on the capital ship.—"The Steamship," September, 1924.

SURFACE CONDENSERS OF THE GINABOUT TYPE.*

With the usual arrangement, in horizontal rows, of the tubes in a surface condenser, a tube in any but the top row is subjected to water dripping on its center, which keeps its whole surface bathed in a hot film; this must reduce its heat-absorbing power. Mr. Ginabout arranges the tubes in sloping planes at such an angle that the drip from one tube falls on or near to the side of the tube below. Instead of passing over 180 degrees on each side, the hot water thus runs over little more than one-fourth of the surface, leaving nearly three-quarters of each tube comparatively dry. Passages between the groups of tubes are arranged so that the steam meets the dry side of the tubes, any air escaping past their wetter sides. It is found that no more than 60 to 70 per cent of the usual cooling surface is required with this system. Alternately, hotter circulating water may be used with a given vacuum if the usual area of cooling surface is provided.—"Gén. Civ."—"Mechanical World," August 15, 1924.

* Inst. C. E. Foreign Abstracts.

SHERARDIZING; ITS INCEPTION AND DEVELOPMENT.

By SHERARD COWPER-COLES.

The process of Sherardizing, or vapor galvanizing, was discovered in the year 1900, and consists of heating the metal to be coated with zinc in a closed receptacle with zinc dust to a temperature of about 600 degrees F., a temperature below the melting point of zinc. Zinc dust is often confused with zinc oxide, but they are very different products and the process will not work with zinc oxide (ZnO). The zinc powder used in the process is the zinc dust of commerce, and consists of very finely divided zinc coated with zinc oxide (the metallic contents average about 85 per cent and zinc oxide 15 per cent). It is obtained during the process of distilling zinc from its ores. The specific gravity of zinc dust is 6.896. When tightly packed a cubic foot weighs 200 pounds., and a ton of zinc dust occupies 11.2 cubic feet. When examined under the microscope it is seen to contain small bright metallic beads distributed through the dust. One of the peculiar properties of zinc dust is that it cannot be melted or reduced to the metallic form under ordinary conditions, even when heated to a very high temperature under considerable pressure. This property is very advantageous for Sherardizing, as it does away with the risk there might otherwise be of melting the finely divided zinc by overheating the furnace.

As many erroneous and misleading statements have appeared in the Press from time to time, some particulars as to the inception and development may be of interest.

The author was making some experiments in connection with the annealing of iron and tried heating cast iron in a closed receptacle filled with zinc dust, when it was observed that the iron became coated with a fine coating of metallic zinc, and subsequent examination showed the zinc had not only coated the iron but had alloyed with it and penetrated into the iron surface. This discovery led to a number of investigations being undertaken, and ultimately a small plant was erected in a workshop at Danvers street, Chelsea. The apparatus used consisted of a wrought-iron cylinder with an opening at the side for inserting the zinc dust and articles to be Sherardized; the cover was made tight by driving in stepped wedges so as to force it against an asbestos packing. The drum was heated by a row of gas burners placed beneath it, and was rotated continuously or intermittently. This trial apparatus proved so successful that a small company was formed, who erected a commercial plant at Willesden. The plant comprised four furnaces capable of taking drums of 8 feet × 2 feet with a cubic capacity of about 2 tons of material at a charge, varying according to the shape and thickness of the articles to be Sherardized. The furnaces were heated by Dowson gas, led by iron pipes to the back of the furnace, and conducted through brick channels, through which the necessary air was drawn, the gas being burnt through large cast-iron rectangular burners placed beneath the revolving drums. The charging of the drums was effected by running the truck on which the drum was placed on to a table one end of which was lowered by means of gearing, so as to tilt the other end into which the zinc dust was charged from an upper floor by means of a chute. The drum was discharged, after the Sherardizing operation had been carried out, over an iron grating, allowing the zinc dust to fall into a chamber below from which it was raised by means of a chain elevator to the floor above. When the drum was charged with zinc dust and the articles to be Sherardized, it was brought into a horizontal position, the air exhausted, and the drum and under-carriage run along the rails until it arrived in front of the furnaces. The drum was then lifted on to a furnace under-carriage, the object being to effect a saving in the first cost of the under-carriages and to save waste of heat. The drum was then pushed into the furnace, the door lowered, and

the furnace heated up to the desired temperature. The heat was regulated in accordance with the readings of a thermometer which was placed in a horizontal iron tube inside the drum, through one of the trunnions which was hollow. When the drum had been in the furnace a sufficient time to give the desired result the door was raised and the drum and under-carriage withdrawn, the drum lifted on to another under-carriage and run into an open yard where it was allowed to cool down to a temperature low enough to admit of easy handling of the contents.

It soon became apparent that Sherardizing could be applied to the inlaying and ornamenting of metallic surfaces, producing effects somewhat similar to those obtained by Damaskeening, the chief difference being that the metals are truly united; the designs are not only inlaid but can be raised and the metals blended together, forming a variety of alloys of many colors and tints, and it was found that when desired the inlaying could be made to penetrate through the surface to the back, and that the thickness and depth to which the metals could be inlaid or onlaid could be readily controlled by the operator. The process is capable of being applied to a large variety of metals, and has the advantage that many alloys can be formed in the one operation of baking. The *modus operandi* is as follows:—The article to be treated is coated with a "stopping off" composition—a thick paste of whitening and linseed oil—those portions which are to be inlaid being left exposed. It is then placed in a box containing the zinc powder and baked at a temperature of about 600 degrees F., the depth of inlay depending on the length of time of the "bake," while the intermediate effects of the alloys and colorings are obtained by manipulation of the "stopping off" material. If the "stopping off" composition is made thin in parts the inlaying process will be retarded, and those portions where the "stopping off" material has been entirely removed will be inlaid: as an example, if copper is being inlaid with zinc, when the "stopping off" is thin the copper will be converted into brass, and where the "stopping off" is entirely removed the copper will be inlaid with zinc. The process is inexpensive and can be applied to the finest work, for instance, photographic work can be transferred. The process can be readily and effectively applied to the decoration of frames, table tops, panels, trays, vases, electric light fittings; in fact, to all classes of metal work. Some good specimens of this work can be seen at the South Kensington Museum.

The rights for America were sold, and a parent Company formed to control them. The policy adopted by this Company was exactly the reverse from that adopted by the English Company and proved very successful. The American Company granted licenses to firms on a minimum royalty basis. The result was that Sherardizing soon became an industry of considerable magnitude; almost every large engineering firm taking a license.

The articles which have been Sherardized are freed from all adherent zinc dust, by substituting a perforated lid for the solid one used when Sherardizing and subjecting the drum to a rumbling action in a closed chamber. A considerable economy in zinc dust is thus effected, and the time required for cooling reduced. A drum can be placed directly under this cooling and discharging apparatus so that the zinc dust can be discharged direct into it while warm and be immediately used again.

At the present time there are about one hundred plants in operation in the United States, and a large number on the Continent. The Germans during the War made most of their coinage out of iron which was Sherardized.

The original Sherardizing Patent, which was a controlling one, was taken out in 1901, and therefore has expired, so a number of firms are now using the process for their own special productions, and some firms undertake all classes of work for the trade. A few of the firms employ fancy distinctive names, such as "Thermozinc," "Galeco," "Zinc Alloy."

Calorizing is a development of the Sherardizing process, utilizing aluminium powder instead of zinc dust and baking at a much higher temperature. The process is found useful for coating ironwork which is subjected to high temperatures as the aluminium coating prevents the rapid oxidation and scaling of the iron or steel and increases its life several fold.—"Engineering," August 29, 1924.

ITALIAN THEORIES OF SUBMARINE CONSTRUCTION.

Admiral Thaon di Revel's program of naval construction allowed for 16 submarines, of a class which was not specified; and these boats, if added to the four already being built, would give the country a submarine fleet of 63 units in 1928. For various reasons the building of these 16 additional boats has not yet been authorized; and, while the matter is still in suspense, two very interesting contributions to the theory of submarine design have been published by Italian naval officers. Signor G. Sechi, in his "reflections upon submarines," insists upon their intrinsic limitations. He admits that the larger type of submarine cruiser may do great damage to commerce, but argues that it is only by leaving her submarine characteristics behind that she does so. If a submarine is to operate successfully against commerce on the ocean highways, at a great distance from her base, she must rely upon gunfire for destroying her quarry; this means that submarines of this kind will more and more approximate to surface cruisers which have the power to submerge. There is an important corollary to this. No submarine cruiser will ever have a total combatant power anything like equal to that of a surface cruiser. The larger types therefore tend to become hybrids both in conception and design. They will be bad submarines owing to their size; they will be bad surface cruisers, because their gun power will necessarily be low, and their range-finding apparatus will be handicapped by its proximity to the water line. In all this Signor Sechi practically repeats the brilliant arguments of Captain Castex, the author of "Synthese de la guerre sousmarine"; but he applies his conclusions to the Italian building program. Powers with relatively low financial resources ought to concentrate upon torpedo-carrying submarines of moderate displacement, and use them as a first line coastal defense. It is a pity that Signor Sechi did not push his inquiries a little further to examine a question which is of the first importance to naval powers of medium strength such as Italy. The question is, simply, the degree of protection which ocean-going submarines can give to convoys of merchantmen which are likely to be attacked by surface cruisers and battle cruisers. If a type of submarine can be devised, which will safely escort fleets of merchantmen to harbor in the face of attacks from powerful surface ships, then, some of the most elementary assumptions of naval strategy will need revision. It should also be said that many British officers believe today that submarines can give convoys of merchantmen the necessary protection, either by delaying the movements of the attacking force, or by so damaging it that the loss or dispersal of the merchant fleet is a good exchange in military values.

The second contribution to the subject comes from Captain de Feo, who deals with the question by suggesting a special design of submarine, and by discussing its merits and defects. An "O" type submarine—which is the name he gives to it—would displace 1020 tons, and be armed with two 4-inch anti-aircraft guns, and eight 21-inch torpedo tubes, for which she would carry 450 projectiles and 12 torpedoes. She would be fitted as a mine-layer with a store of 18 mines; but if, for any reason, it were decided to arm her simply as a torpedo carrier—that is, if the mines were removed—the number of torpedoes could be raised to 18. Her normal fuel capacity would be 71 tons, but this could be nearly doubled. Her radius of action moving at

economical speed, which is 8 knots, and with a maximum load of oil, would be 9000 miles. Her highest speed would be 19 knots, and she could maintain it for 2400 miles with a full cargo of fuel.

The peculiarity of the design is that all the flooding chambers are placed right ahead and right astern, and that the vital organs—motors, living quarters, batteries, etc.—are grouped between the two. The result of this is that only this central part of the submarine need be built to resist pressure. Captain de Feo claims that by thus reducing the specially strengthened part of the hull he economizes displacement which can be devoted to combatant power. The advantages, taken in order, are as follows: (*a*) that, given equality of tonnage and resistance to pressure, the "O" type submarine has a lighter hull than any other type; (*b*) that, by its design, the "O" type submarine is 50 per cent less vulnerable than any other. On this point the designer's argument is that a submarine must protect herself principally from mines and depth charges, as superior gunfire can generally be avoided by diving.

Dealing with each of these dangers in turn, Captain de Feo argues that, when a submarine passes through minefields, she generally strikes them in her fore part; so that, by placing the flooding chambers right forward, the design interposes a non-resisting section—which can be damaged without losing the submarine itself—between the danger and the vital organs. Similarly, depth charges will only do irreparable damage if they strike the central, resisting, part of the hull, which means that the vulnerable target is reduced by half. (*c*) That it is not necessary to put ballast along the keel line of an "O" type submarine, as it maintains sufficient metacentric height without it. Captain de Feo's proof of this point is necessarily long and intricate: he claims, however, that his submarine would have a metacentric height of fifteen inches when submerged, although the engines and batteries are higher placed than in other types. (*d*) That the "O" type submarine would be able to increase her speed and fighting power at a greater ratio for every ton of displacement than any other class. The first part of the claim follows from the economy of weight which is at the basis of the design, the second needs explanation. If a submarine's metacentric height is to be maintained, the weight of additional armament, being always placed above the center line, has to be compensated by placing additional weights below it. These additional weights are paid for by lower battery power, and less resistance to pressure. The British M type, for instance, has paid for its armament of 12-inch guns by only being able to resist pressure up to 100 feet. Captain de Feo claims that an "O" type submarine, with its great metacentric height, could mount a 12-inch gun without adding compensating weights along the keel line, which means that she could do so without loss of speed or capacity to submerge deeply. Captain de Feo might, with advantage, have been slightly more explicit on this point. He means presumably that an "O" type submarine can mount a 12-inch gun without loss of anything but metacentric height, that is, of stability: he would, however, have done well to support his contention by calculation. (*e*) That the centralizing of all vital organs and maneuvering mechanism is an advance on all other types. The sub-divisions and compartments of all submarines of the "double hull" type make it difficult for the commanding officer to communicate his orders rapidly and effectively to all parts of his vessel. (*f*) During the war it was found that the slightest leak in the accumulator chambers created poisonous gases, as the water had immediate access to batteries. By being able to place the accumulators relatively high up, as he can do owing to the high stability of his vessel, Captain de Feo claims that he has secured a great factor of safety. (*g*) By keeping the flooding chambers at each extremity, the ship maintains a big transversal metacentric height whether she is on the surface or submerged. The designer's other claims more or less follow from those already enumerated, and need not be described.

The criticisms which could justly be levelled against the design are that the "O" type would lose protective strength by having only one hull; that the chambers at either extremity would not be easy to flood; and that the position of the accumulators is not satisfactory in that it leaves them insufficiently isolated.

Captain de Feo's design seems to be the outcome of a theoretical principle which developed during the war. In 1918 and 1919 it was generally stated that submarines, being slow, without vision and very vulnerable, could not possibly survive the means of destroying them which had been devised during the course of the war. There is a great deal in this contention; and it has to be admitted that, in spite of its mechanical development, the latest German U-boat had not really overcome the inherent limitations of the first Holland boats. Submarines will only make real advance if designers strive earnestly to get away from the "congenital defects" of the existing type. Captain de Feo doubtless regards his "O" boat as a first contribution.—"Engineering," August 15, 1924.

STEAM AT 1200 DEGREES F. IN A UNIQUE POWER PLANT.

Steam pipes glow red at 750 degrees F. when viewed in the dark. The diminutive power plant shown herewith utilizes steam at full cherry red heat. Although this boiler and turbine are used for propelling torpedoes, there are features which suggest possibilities for more economical stationary practice.

A twenty-year advance over present steam-engineering practice is represented by the working temperature of 1200 degrees F., with past progress as a measure. The difficulties of generating and utilizing the steam under such conditions can be realized from the fact that iron loses its strength to such a degree at red heat that the blacksmith can readily shape it with a hammer. Twelve hundred degrees, which is a cherry red, verges on this condition. A pressure of 350 pounds to the square inch is here applied, which makes these steam conditions all the more remarkable.

Although there are many ways of increasing plant economy, elevating the temperature of steam offers greater theoretical possibilities than any other means. It raises the possible limit to which engineers may strive as a goal. Four hundred degrees increase, as here would obtain, means that the Carnot or limiting efficiency is raised from about 55 to 66 per cent.

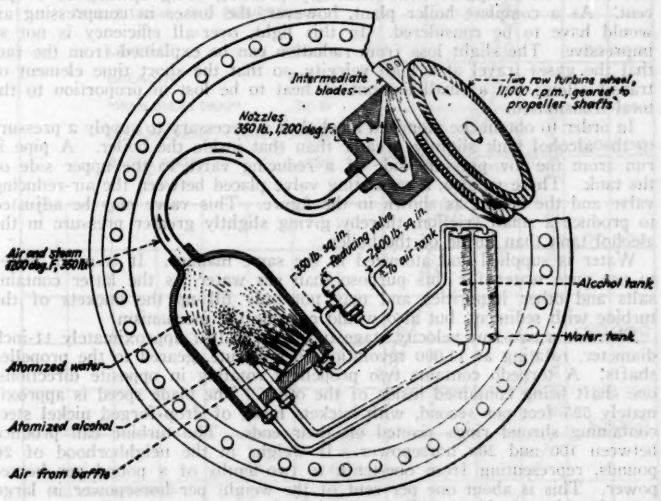
Raising the steam temperatures, however, is probably one of the most difficult means of improving plant efficiency. Ordinary metals become much weaker and what is more important, lose their elasticity at an even greater rate. New alloys must be developed to meet such conditions. It is not to be wondered at, therefore, that progress in this direction has been slow. The average rate of increase of steam temperature during the last ten years is about fifteen degrees per year, for land practice. In the future the rate of increase will probably be more rapid than in the past.

The torpedo power plant, unlike stationary plants, operates for short time intervals only. The average run is in the neighborhood of five minutes, and the total life of the turbine, without repairs, from fifty to two hundred hours. It is therefore far removed from conditions of stationary practice, where long uninterrupted operation is required. A turbine, however, for 1200 degrees F., 350 pounds, supplied by a boiler where fuel and water are intimately mixed, at high combustion efficiency, presents a new phase in regard to the possibilities of future progress.

Torpedoes were originally driven by compressed air. It was found that heating the air prevented freezing temperatures, incident to the expansion from heavy pressures to a low working pressure. Not only this, but the

oxygen contained in a pound of air, if properly burned, would make it considerably more powerful and would also heat the air to 1600 degrees F., which is much too high for ordinary materials.

In order to conserve the power represented by the difference in temperature of 1200 degrees, which is usable, and 1600, which cannot be used, water is sprayed into the mixture after combustion has taken place. This is in



AIR, FUEL AND WATER ARE MIXED IN ONE CHAMBER AND SUPPLIED TO TURBINE AT 1200 DEGREES F.

Inclined Positions of Boiler and Turbine Are Unlike Land Practice. Although Parts at High Temperature Are Submerged in Salt Water, Radiation Loss Is Very Low on Account of High Gas Velocities.

sufficient quantity to bring the temperature down to 1100 or 1200 degrees. Steam of 350-pound pressure therefore will be superheated about 700 degrees. The air will thus have its working power increased about 250 per cent.

The simplicity and effectiveness of the plan in general can be seen in the figure. Air is stored at approximately 2600 to 2800 pounds per square inch and is fed through a reducing valve to the boiler, which it enters at approximately 350 pounds. The air is admitted through holes in the baffle plate, so that it is diffused in the boiler.

Alcohol is atomized and injected into this air, allowing efficient combustion to take place. After this, atomized water is sprayed into the gases, so as to control the temperature. The boiler is comparatively small, being approximately 8 inches long, with the air, alcohol and water all injected at the bottom. The heated gases escape at the top, as indicated by the arrows. Valves are arranged so that, when igniting, the alcohol reaches the boiler before the water.

The construction of a torpedo is such that the boiler and piping are normally submerged in salt water. Efficiency is not a special object of this device, the boiler not even being lined with refractory material. There is, of course, good reason for this, inasmuch as the cooling action of the water tends to make the material of the piping better able to resist this temperature.

In spite of the fact that a difference of 1000 degrees may exist between the heated gases and the surrounding water, the efficiency of combustion is not bad when compared with stationary practice, being upward of 96 per cent. As a complete boiler plant, however, the losses in compressing air would have to be considered. In this light, over-all efficiency is not so impressive. The slight loss from radiation can be explained from the fact that the gases travel at a high velocity, so that the short time element of transit allows but a small amount of heat to be lost in proportion to the total transmitted.

In order to obtain the atomized alcohol, it is necessary to supply a pressure to the alcohol tank slightly greater than that inside the boiler. A pipe is run from the low-pressure side of a reducing valve to the upper side of the tank. There is, also, a regulating valve placed between the air-reducing valve and the boiler, as shown in the figure. This valve may be adjusted to produce a small friction, thereby giving slightly greater pressure in the alcohol tank than inside of the boiler.

Water is supplied and atomized in the same manner. It is much better to use pure water for this purpose than sea water, as the latter contains salts and other impurities and may not only fill up the buckets of the turbine with sediment, but also would corrode the mechanism.

There is a two-row velocity stage turbine wheel of approximately 11-inch diameter, rotating at 11,000 revolutions per minute, geared to the propeller shafts. A torpedo contains two propellers rotating in opposite directions, one shaft being contained inside of the other. The blade speed is approximately 525 feet per second, with buckets made of drop-forged nickel steel containing shroud rings riveted on their ends. The turbine can produce between 100 and 200 horsepower. It weighs in the neighborhood of 20 pounds, representing from one-tenth to two-tenths of a pound per horsepower. This is about one per cent of the weight per horsepower in large central-station machines.—"Power," September 9, 1924.

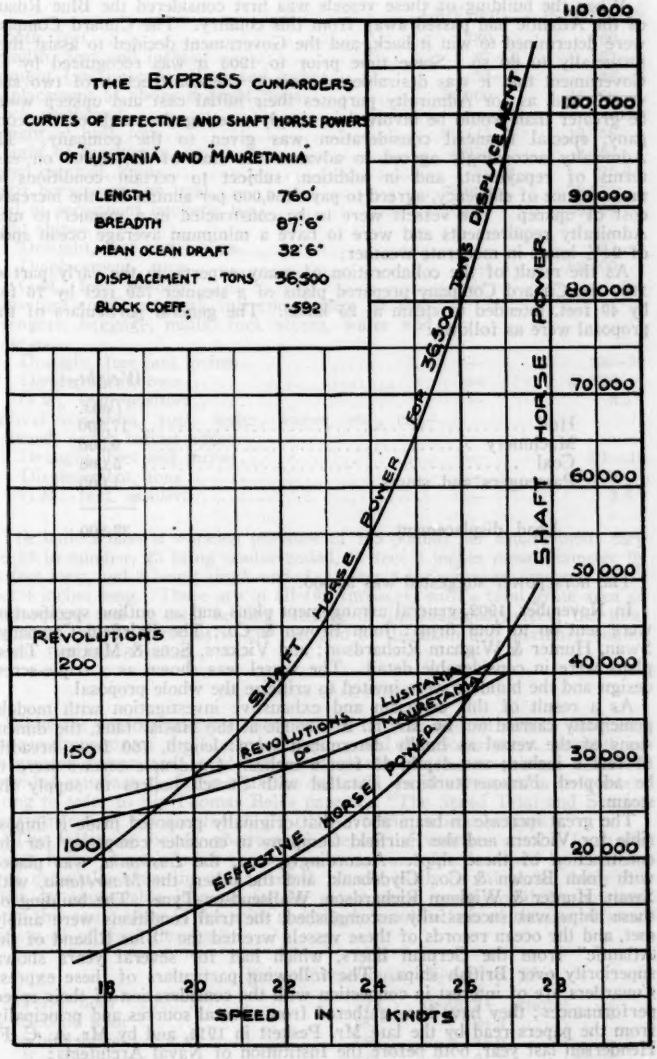
WHAT THE "BLUE RIBAND OF THE ATLANTIC" COSTS.

Some Interesting General and Technical Details of the Cost of Running Crack Atlantic Passenger Liners.

Intense interest has been aroused in the recent record-breaking performances of the Cunarder *Mauretania*, which vessel has been excelling her previous excellent work, due largely to the installation of oil-burning apparatus, which gets over difficulties of the maintenance of steam.

In order that those interested in these performances may arrive at some idea of the expenditure in power which these necessitate, the accompanying diagram has been prepared from the published records of the *Lusitania* and *Mauretania*. The enormous cost attending the great speed achieved can also be judged from a casual glance at the curve of powers.

In view of the unrivalled position among the Atlantic greyhounds, which the *Mauretania* has maintained since her construction, which position was challenged by one vessel only, viz., her ill-fated sister ship, the *Lusitania*, it would not be amiss to consider the purpose of the construction of these ships, the problems which were met and successfully overcome, and the wonderful performances attained by both vessels.


THE EXPRESS CUNARDERS
CURVES OF EFFECTIVE AND SHAFT HORSE POWERS
OF "LUSITANIA AND "MAURETANIA"
LENGTH 760'
BREADTH 87'6"
MEAN OCEAN DRAFT 32'6"
DISPLACEMENT - TONS 36,300
BLOCK COEFF. ·592
SHAFT HORSE POWER FOR 36,300 TONS DISPLACEMENT
SHAFT HORSE POWER
110,000
100,000
90,000
80,000
70,000
60,000
50,000
40,000
30,000
20,000
REVOLUTIONS
200
150
100
SHAFT HORSE POWER
REVOLUTIONS
EFFECTIVE HORSE POWER
LUSITANIA
MAURETANIA
18
20
22
24
26
28
SPEED IN KNOTS

When the building of these vessels was first considered the Blue Riband of the Atlantic had passed away from this country. The Cunard Company were determined to win it back, and the Government decided to assist them financially to do so. Some time prior to 1903 it was recognized by the Government that it was desirable to promote the construction of two such vessels and, as for Admiralty purposes their initial cost and upkeep would be greater than would be involved in simply meeting the needs of the company, special financial consideration was given to the company. The Admiralty accordingly agreed to advance the sum of £2,600,000 on easy terms of repayment, and in addition, subject to certain conditions of maintenance of efficiency, agreed to pay £150,000 per annum for the increased cost of upkeep. The vessels were to be constructed in a manner to meet Admiralty requirements and were to have a minimum average ocean speed of 24½ knots in moderate weather.

As the result of the collaboration of many experts, in the early part of 1902 the Cunard Company prepared plans of a steamer 750 feet by 76 feet by 49 feet, intended to steam at 25 knots. The general particulars of this proposal were as follows:

	Weight—
	Tons.
Hull	17,500
Machinery	9,300
Coal	5,000
Passengers and stores	1,000
Load displacement	32,800

The horsepower suggested was 55,000.

In November, 1902, general arrangement plans and an outline specification were sent on to four firms: John Brown & Co.; The Fairfield Company; Swan, Hunter & Wigham Richardson; and Vickers, Sons & Maxim. These plans were in considerable detail. The vessel was shown as a triple-screw design and the builders were invited to criticize the whole proposal.

As a result of this criticism and exhaustive investigation with models, principally carried out by Mr. R. E. Froude at the Haslar tank, the dimensions of the vessel as finally determined were: length, 760 feet; breadth, 87 feet 6 inches; and depth, 60 feet 6 inches. Quadruple screws were to be adopted, Parsons turbines installed with Scotch boilers to supply the steam.

The great increase in beam above that originally proposed made it impossible for Vickers and the Fairfield Company to consider competing for the construction of these ships. Accordingly, one, the *Lusitania*, was placed with John Brown & Co., Clydebank, and the other, the *Mauretania*, with Swan, Hunter & Wigham Richardson, Wallsend-on-Tyne. The building of these ships was successfully accomplished, the trial conditions were amply met, and the ocean records of these vessels wrested the "Blue Riband of the Atlantic" from the German liners, which had for several years shown superiority over British ships. The following particulars of these express Cunarders are of interest in connection with the consideration of their speed performances; they have been gathered from several sources and principally from the papers read by the late Mr. Peskett in 1914, and by Mr. A. C. F. Henderson last year, both before the Institution of Naval Architects:

	Lusitania.	*Mauretania.*
Length, feet	760	760
Breadth, feet and inches	87—6	87—6
Draught (mean ocean), feet and inches	32—6	32—6
Displacement at above draught, tons	36,500	36,500
Block coefficient	.592	.592
Weight of hull, tons	20,145	19,758
Weight of machinery, tons	9,936	9,402
Speed on trial, knots	25.62	26.04
Shaft horsepower on trial	76,000	78,275
Coal in permanent bunkers, tons	6,029	6,354
Light ship—		
Draught, feet and inches	—	27—2
Displacement, tons	—	29,600
G.M., feet, positive	—	.3
Departure condition, with full complement of passengers, baggage, mails, fuel, stores, water and cargo—		
Draught, feet and inches	—	35—10
Displacement, tons	—	39,370
G.M., feet, positive	—	3.5
Arrival condition, fuel, water, stores, etc., consumed, water ballast added—		
Draught, feet and inches	—	31—10
Displacement, tons	—	33,958
G.M., feet, positive	—	1.4

The boilers have a working pressure of 195 pounds per square inch; they are 25 in number, 23 being double-ended, 17 feet 6 inches mean diameter by 22 feet long, and 2 being single-ended, 17 feet 6 inches mean diameter by 11 feet 4 inches long. There are in all 192 furnaces, with a total grate area of 4048 square feet; a total heating surface of 158,352 square feet; the total length of the boiler rooms is 336 feet and of the main and auxiliary engine rooms 149 feet 8 inches.

OIL VERSUS COAL FUEL.

While the performances of these vessels on trial and on service fully met the requirements of the Cunard Company and the Government, possibilities of greater power and speed were undoubtedly in the machinery installed, provided satisfactory firing could be ensured. In this connection it is interesting to refer to Sir Thomas Bell's paper on "The Speed Trial and Service Performance of the *Lusitania*" (T.I.N.A., 1908), where he says:

"The following hourly abstract of one of the watches on the *Lusitania* brings home to one's mind the loss in steam and speed caused by cleaning fires, especially when the coal is small. It can easily be calculated from this what an appreciable increase in the ship's mean speed could be obtained from this cause alone if the price and supply would admit of the use of some system of oil fuel burning."

	Mean Revolutions.	*Speed of Ship.*	
First hour	178	about 24 knots	24.15
Second hour	181	24.3	
Third hour	186	25.0	25.05
Fourth hour	187	25.1	
Mean for watch	183	24.6	

Unfortunately, the *Lusitania* was lost before it was commercially possible to install oil-fuel apparatus, but the recent records of the *Mauretania* indicate the advantages to be gained by such, as indicated by Sir Thomas Bell in the paper quoted above.

PERFORMANCES OF THE "LUSITANIA."

While the *Mauretania* has naturally loomed large before the public eye, the achievements of the *Lusitania* were no less meritorious, as the following record of her out and home voyages for the year from January, 1911, to January, 1912, shows:

Westward.					*Eastward.*				
Date.	*Time.*				*Date.*	*Time.*			
1911.	*Dys.*	*Hrs.*	*Mins.*	*Speed.*	1911.	*Dys.*	*Hrs.*	*Mins.*	*Speed.*
Jan. 8....	4	13	35	25.37	Jan. 18....	4	18	40	25.57
Jan. 29....	5	0	48	23.92	Feb. 8....	4	19	29	25.39
Feb. 19....	5	19	0	22.22	March 1....	4	20	28	25.18
March 12....	4	18	10	25.31	March 22....	5	0	24	24.35
April 9....	4	17	58	25.35	April 19....	4	20	42	25.13
April 30....	4	18	42	25.19	May 10....	4	21	37	24.93
May 21....	4	18	0	25.36	May 31....	4	23	42	24.51
June 11....	4	19	59	24.90	June 21....	4	23	31	24.53
July 9....	4	17	10	25.53	July 19....	4	21	14	25.04
July 30....	4	16	38	25.65	Aug. 9....	5	0	58	24.24
Aug. 28....	4	18	37	24.25	Sept. 4....	4	20	50	24.06
Sept. 12....	4	16	18	24.76	Sept. 20....	4	22	55	23.68
Oct. 8....	4	14	51	25.08	Oct. 18....	4	19	41	24.26
Oct. 29....	4	18	0	24.39	Nov. 8....	4	20	30	24.09
Nov. 19....	4	19	30	24.07	Nov. 29....	4	23	25	23.50
Dec. 10....	4	20	40	25.35	Dec. 17....	4	23	40	23.46
Dec. 31....	4	19	18	24.11	1912.				
					Jan. 10....	5	6	53	22.12

In all, 34 voyages across the Atlantic in one year at an average speed of 24.55 knots. This is a wonderful record, not only meritorious as far as reliability of ship and machinery is concerned, but in the quickness of turn-round when coal bunkers had to be filled and stores and baggage unladen and laden.

THE "MAURETANIA'S" NEW RECORDS.

The *Mauretania* recently accomplished the trip from Cherbourg to Ambrose Light in 5 days 3 hours 20 minutes, at an average speed of 25.6 knots; the voyage from New York to Cherbourg in 5 days 1 hour 49 minutes, at an average speed of 26.25 knots; also in an eastward day of 23 hours 9 minutes she steamed 626 miles, an average speed of 27.04 knots. If the diagram be referred to it will be seen that this last speed represents about 96,000 S.H.P. The power curve has been derived from the results given in Sir Thomas Bell's paper; the effective horsepower curve was given by Mr. Peskett in his 1914 paper.

It is to be observed that the curve of horsepowers is obtained from trial results. Accordingly, it is possible that still greater power than that indicated will be required at sea.

The late Sir Wm. White gave as his opinion that the size and power of these liners was admirably suited to maintain high speeds in all weathers. From recent investigations it appears that the amount of power necessary to overcome wind and wave resistance may be a relatively small portion

of the total power available in the case of a high-powered vessel. The records of these vessels show this to be the case as they maintain their speeds and perform their voyages with the regularity of express trains.

One or two points in connection with these records deserve attention. The importance of having ample boiler power in turbine work is indicated by these results. Originally, when the *Mauretania* and *Lusitania* were coal-burning vessels, a margin of about one-fifth the boiler power was assumed out of action through cleaning and ineffective stoking. With the introduction of oil fuel, a large increase in steam supply is available, which increase, apparently, can be satisfactorily used by the turbines.

The suitability of direct-driven turbines for high-speed craft of this type is also indicated. It is possible that had gearing been thought of at the time when these vessels were contemplated smaller turbine units with single-reduction gearing might have been adopted. It is questionable whether the regularity of service and the freedom from trouble which the above table of the *Lusitania's* performances indicate would have been maintained had the machinery been less simple than that fitted.

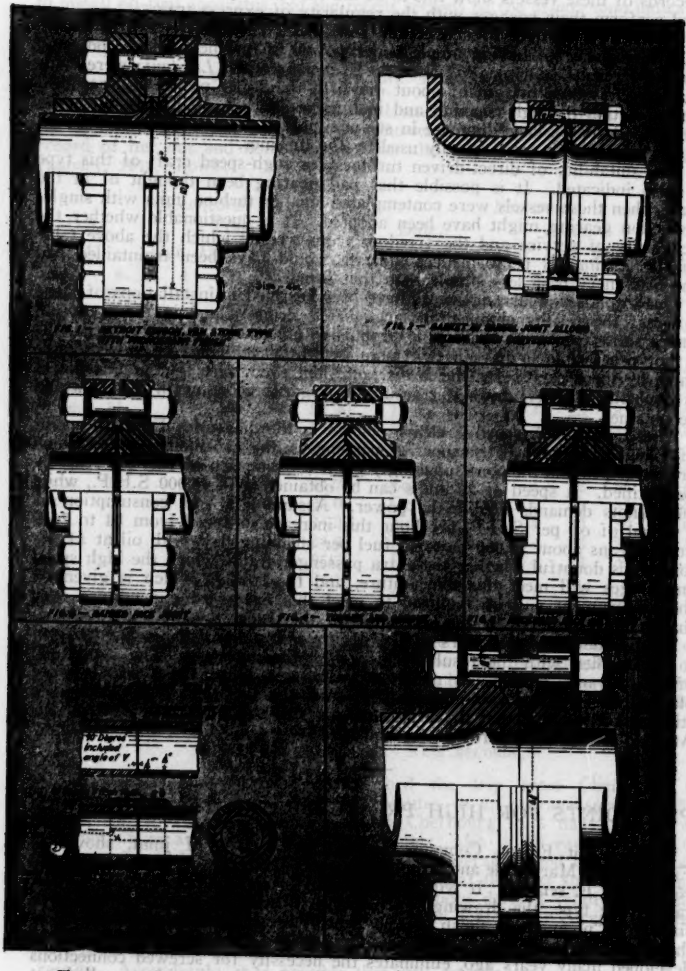
Quadruple-screw arrangements have been justified in installations of large powers; the results obtained in large battle cruisers and in these express liners show that quite reasonable propulsive efficiencies can be secured. It is not practicable to have a twin arrangement developing the powers indicated by these curves for the *Mauretania*, but even were that practicable, it is doubtful whether any better results as regards economy in power would be obtained.

The cost of high speed is clearly shown in the diagram. From the point of view of economical transport of passengers, the enormous powers which are developed and the correspondingly great consumption of fuel can scarcely be justified. A speed of 23 knots can be obtained with 45,000 S.H.P., while 26½ knots demands double the power. Assuming a fuel consumption of 1 pound of oil per S.H.P. per hour this increase in speed from 23 to 26½ knots means about £2,000 more in fuel per steaming day, with oil at £4 per ton. It is doubtful whether the extra passenger traffic which the high speed encourages will meet this expenditure, and the subsidy which is given by the Government will be required to meet this excess of expenditure over that actually required by the necessities of the trade.

The builders and designers of the *Mauretania* and *Lusitania* have reason to be satisfied with the results achieved by these vessels. The magnitude of the advance which was made 20 years ago, when these vessels were planned and laid down, is made evident from the fact that their records are still unexcelled, and that the *Mauretania* is still the foremost vessel on the Atlantic service as regards power and speed.

PIPE JOINTS FOR HIGH PRESSURES AND TEMPERATURES.

The Detroit Edison Company has utilized a type of joint, shown in Fig. 1, at the Marysville and Trenton Channel plants, designed for 400-pound 750-degree F. service. It will be seen that this is the familiar Van Stone raised-face type, the drawing representing to scale a 4-inch nominal pipe size, with "phonograph finish." The Detroit Edison standards do not allow threaded flanges in 4-inch pipe sizes or larger. The Van Stone principle, developed many years ago, eliminates the necessity for screwed connections between pipe and flange. Ends of the adjoining pipes are here rolled out at right angles, forming flanges solid with the pipes, and loose collars assembled as shown, bolt together so as to hold the pipe ends rigidly together. These joints may be machined if desired for raised face, as usually is the case as well as tongue and groove, or any other design of



TYPES OF JOINTS FOR HIGH PRESSURE AND TEMPERATURE SERVICE.

contacting surfaces. Reinforcing rings ordinarily are welded to the flanges before machining, so that the thickness of metal when finished, will not be less than that of the pipe itself.

The clearances and curvatures of the bolted collars in the figure are arranged so as to give maximum pressure in the extreme outside region of the joint. This is for the purpose of concentrating the compressive force around the outside edge of the flanges so that the tendency will be to produce a wedge-shaped gasket space, with the greatest width of the section toward the interior of the pipe. Any movement of the gasket, therefore, due to a tendency to blow out, will tend to pack it more tightly in the wedge-shaped space. Dimensions correspond closely to the 600-pound A.E.S.C. tentative standards, published in the December 4, 1923, and March 18, 1924, issues.

The "phonograph finish," or roughening of the raised faces, gives an additional means of confining the gasket, which is one of long-fibered asbestos. Thus, the advantages of convenience in erecting and replacing the raised-type face of joint is combined with a greater degree of relative steam tightness than the raised face otherwise possesses. This roughening is accomplished on each face by cutting grooves about $\frac{1}{64}$ inch deep with a round-nose tool. These may be in the form of a spiral or else a series of circles eccentric with the pipe. The grooves are spaced about 32 to the inch. This joint has been thoroughly successful at the Marysville plant, after an extensive test in 300-pound 700-degree service. The brief trial at the Trenton Channel Station with 420-pound 725-degree steam, although satisfactory to date, has not been sufficient for passing judgment.

The usual raised-face joint, Fig. 3, which fulfills the requirements of moderate pressures so well, is less able to withstand high pressure than the tongue-and-groove, Fig. 4, or the recessed face, Fig. 5. Tests reported November 17, 1922, to the N.E.L.A. Prime Movers Committee showed that with the same gasket pressures and materials it was possible to blow the gasket out of a raised-faced pipe joint at a lower pressure than would cause a leak in the tongue-and-groove type. There was no instance in these tests where the former would withstand one-half the pressure of the latter under similar conditions. Gasket pressures in the neighborhood of ten to twelve times the steam pressures have been highly successful in process work, where temperatures and pressures relatively are higher than those of the present maximum steam conditions.

Experience with gasket materials at maximum steam temperatures and pressures has been quite conflicting. Asbestos may solve the problem very well at one installation, yet in another of similar conditions metallic gaskets would be preferred. In process work with temperature around 800 to 900 degrees, narrow metallic rings at high gasket pressure here frequently found favor.

The joint in Fig. 5 may be satisfactory as regards relative leakage, but shares with the tongue-and-groove joint the disadvantage of difficulty in erecting and replacing. In each case the pipe line must be separated sufficiently at the joint, so that the projecting portions of one flange may be withdrawn from the recesses in the other.

The Sargol pipe joint, Figs. 2 and 8, has been developed for the purpose of securing a greater degree of tightness than the ordinary raised-face Van Stone joint allows. It will be seen that these joints are very similar to Fig. 1. Provision, however, is made for welding both of the pipe ends together so that a gasket is not required. In replacing such joints, chipping and welding are necessary procedures. One of the first stations to operate in the neighborhood of 700 degrees 300 pounds, was the Springdale plant of the West Penn Power Co., which adopted the Sargol type. Where special adjusting or replacement of pipe sections was necessary, a gasket could be placed between the flanges so that welding could then be postponed until a convenient opportunity arrived.

Welded-type joints have not so far been utilized to any noticeable extent for maximum steam conditions, although for the fastening of nozzles and miscellaneous work, welding occupies an important position. The joint, as in Fig. 6, should be sufficient for 4-inch pipe at 750-degree 400-pound service except for the fact that it lacks the rigidity of a bolted connection, and therefore, under conditions imposed by pipe flexure, abnormal circumstances, etc., the reinforced weld, Fig. 7, is preferable, which weighs and costs less than a bolted type. These are of interest as a future possibility, when temperatures and pressures have become greatly increased and bolted joints are at a greater disadvantage.

Experience with small pipes in the neighborhood of 1 or 2 inches indicates that for high pressures, raised-face flanges screwed to the pipe and fastened by an additional welding around the threads to prevent leaks, give satisfactory service.

Six hundred pound pressure will be utilized at the Philo station of the Ohio Power Co. The joints here employed are of the Sargol type, as in Fig. 8. Each 1-inch bolt is turned to approximately root diameter, so that if any stretching occurs, it would be equalized over the entire length of the shank, and thus reduce the tendency to break in the thread. This corresponds to the A.E.S.C. standard for 900 pounds being drawn to scale for 4-inch nominal size pipe.—"Power," September 9, 1924.

HIGH SPEEDS AT SEA.

When landsmen speak of an aeroplane as having travelled at 150 miles an hour, or a railway train at 70 miles an hour, the significance of such figures is at once apparent to everybody, sailors included; but when the sailor, on his part, refers with pride to the number of knots logged by some swift ship in the course of a day's run, comparatively few landsmen are able to grasp the full meaning of the achievement. Notable speed performances at sea do not, therefore, impress the public's imagination quite so deeply as similar records established on land or in the air, where the velocity is measured in miles. True, the difference between an Admiralty knot and a statute mile—6080 feet and 5280 feet respectively—is not very great, but it becomes considerable when we are dealing with long runs or very high speeds, as the case may be. Thus, the news that the *Mauretania*, on her latest homeward passage, covered the 3198 miles between Ambrose lightship and Cherbourg at an average speed of 26¼ knots, means that the liner maintained throughout the voyage a speed of more than 30 miles an hour. It should not be overlooked that this wonderful voyage was made by a ship which is now in her seventeenth year. Except that her boilers have been converted for oil fuel and her turbines partly re-bladed, the *Mauretania* remains unaltered. Since her completion she has been almost constantly in service, and probably has a greater mileage to her credit than any other vessel afloat; yet today, when she is already nearing the end of that twenty-year period of "useful life" which is assigned to the average passenger liner, we find her easily surpassing the highest speed she made on her earliest voyages. We can recall no finer example of the British genius for shipbuilding and marine engineering. During the last five hours of the passage to Cherbourg her speed averaged 28¼ knots, or more than 32½ miles an hour. That she could do even better, if it were worth while to force the machinery, was shown by the first run made after the re-blading of the turbines, when for a time she was moving through the water at the extraordinary rate of 35½ miles an hour. The only other vessels of large tonnage able to travel at such a speed are the British battle-cruisers *Hood, Renown* and *Repulse;* but whereas they are rarely driven at anything like full power, the *Mauretania* has been running for many years on a time-table designed with a view to

accomplishing the swiftest Transatlantic crossing, and her machinery is, therefore, normally working at high pressure. We understand that she has averaged 25½ knots for twenty-seven consecutive voyages across the Atlantic, during which she must have encountered many varieties of weather —good, bad and indifferent. In this respect she has more than fulfilled the terms of the agreement between the Government and the Cunard Steamship Company in 1904, under which the *Mauretania* and her ill-starred sister *Lusitania* were built, and which stipulated for an average sea speed in moderate weather of not less than 24½ knots.

That the *Mauretania* represents the last word in high-speed ocean travel, no engineer or naval architect imagines for a moment. While it might be difficult, perhaps impossible, to improve on the lines of the hull, which were modelled with unusual care after an exhaustive series of tank experiments, turbine machinery of higher power and increased efficiency could be built today within the limits of weight and space available for the original plant. From the engineer's point of view it is a matter for regret that no demand should have arisen for the construction of an ocean liner to excel the *Mauretania* in speed. But the design of such vessels is governed by considerations other than fast steaming. High speed at sea is an expensive luxury, and unless the cost thereof can be balanced in other ways the shipowner naturally prefers to invest his capital in ships which are more remarkable for economy in fuel consumption than for swiftness of passage. For this reason practically all the ocean liners built since the war have been designed with moderate engine power. The *Berengaria,* though her gross tonnage is 22,000 more than that of the *Mauretania,* has machinery developing approximately the same power, and neither the *Majestic* nor the *Leviathan,* the two largest vessels afloat, are remarkable for very fast steaming. The former, of 56,551 tons gross, has engines of 66,000 shaft horsepower; the American ship, of 54,282 tons gross, has had her machinery modified to such an extent that her present service speed—according to Brassey—is only 21 knots, which is considerably lower than the original German design provided for. It was announced three years ago that the United States Shipping Board was preparing plans for two Atlantic liners of unexampled dimensions and speed. These ships were to be 1000 feet in length and to have a sea speed of 30 knots, which would enable them to make the crossing in four days. This project, however, appears to have fallen through, as everyone familiar with the economics of shipping was convinced it would. Apart from the high initial cost of such vessels, the fuel and maintenance charges would have reached a staggering figure; and it is doubtful whether a full passenger list on every voyage all the year round would have enabled them to be operated at a profit. Taking the twenty-six largest steamers now afloat, we find the mean of their speeds to be no more than 19.3 knots. Only eight are designed for more than 20 knots, and only four for speeds of 24 knots and upward. Clearly, therefore, the demand for high speeds in the mercantile marine is very limited, and it looks as if the *Mauretania's* possession of the blue riband of the Atlantic would remain unchallenged for years to come. It would indeed be remarkable if the completion of twenty years of service found her still the swiftest merchant ship in the world, yet that may well prove to be the case.

In the design of naval vessels economy of operation is of secondary importance, while the tactical and strategic value of high mobility is unquestioned. There has been, in consequence, a steady rise in the speed of such vessels during and since the war. The battle-cruiser *Tiger,* laid down in 1912, has machinery of 85,000 shaft horsepower for a speed of 28 knots. Four years later Sir E. T. d'Eyncourt planned the *Hood,* in which the engine power was increased to 144,000 shaft horsepower, and the legend speed to 31 knots. On the measured mile this great ship attained 32 knots —36.8 miles an hour—in spite of her displacement being 1000 tons above

the normal. No other major ship of any navy has equalled this performance. The construction of battle-cruisers having been arrested by international treaty, the field of development in fast steaming is now confined to light cruisers and destroyers. As regards the former type, the swiftest vessels afloat appear to be those of the U. S. *Omaha* class, one of which, the *Detroit*, has traversed the measured mile at a speed of 35 knots—40.3 miles an hour—with her engines generating 97,375 shaft horsepower. No pre-war cruiser came within seven knots of this performance. The trial runs of our new light cruisers *Emerald* and *Enterprise* will be awaited with interest; they are designed for a speed of 33 knots when in light condition, with their machinery developing 80,000 shaft horsepower. The French cruiser *Lamotte-Picquet*, which is to be put through her paces during the autumn, has engines of 96,000 shaft horsepower for 34 knots, but her builders confidently anticipate that 116,000 shaft horsepower and 36 knots will be realized. While nothing is known about the details of our new "County" cruisers, it will be surprising if they do not equal in engine power the swiftest vessels now building or projected abroad. The world's record for speed in the destroyer class is still held, we believe, by H.M.S. *Tyrian*, which Messrs. Yarrow completed nearly five years ago. On trial this vessel reached the astonishing speed of 39.72 knots, equivalent to 45½ miles an hour. Such a velocity would have seemed incredible to the older generation of marine engineers, but it is quite likely to be surpassed by destroyers now on the stocks or about to be laid down in this country and on the Continent.—"The Engineer," September 5, 1924.

AN OLD CONTROVERSY REVIVED.

One of the earliest controversies in the electrical power industry was whether the transmission and distribution systems of the future should be direct or alternating current. In this country the direct-current system never made much headway except for three-wire low-voltage distribution in congested city districts, in small and medium-sized industrial plants and for railway service. Today even in many of the smaller industrial plants alternating current is being used. With the exception of a few railway systems all electric-power transmission at voltages above six hundred is done by alternating current.

In Europe this has not been so universally true as on the American continent. In the early eighties a system was developed for transmitting direct-current at three thousand volts, and two thousand volt direct-current railway systems were put into operation shortly thereafter in England. In continental Europe Thury developed his series system of direct-current power transmission, and it has been used to some extent for voltages up to one hundred thousand. As this system requires a large number of generators in series to produce the voltage for transmission and the same number of motors in series driving generators to step the voltage down for local distribution, it has never met with favor. In fact, the system is one of transmission and not of distribution. In later years, as the power systems in Europe have expanded, the tendency has been toward the use of alternating current as in this country.

At the British Empire Exposition the exhibiting of the "transverter," a device for converting high-voltage alternating current into direct current or vice versa has, particularly in Europe, done much to revive the old controversy of direct-current versus alternating-current transmission of electric power. This machine, which has been built in sizes up to two thousand kilowatts, is a step-up or a step-down static transformer combined with a number of commutators connected in series, the brushes of which are driven at synchronic speed. The sizes of this unit and the voltage, one hundred thousand, that it has been designed for, have created much interest in its possibilities.

If alternating-current had reached a limit as to transmission voltage and the industry was looking for some means of overcoming the difficulty, there might be more of an incentive to introduce other complications in power-transmission systems. The 220,000-volt alternating-current system will for a number of years to come meet all our requirements so far as long-distance transmission is concerned. Although such high voltage lines have characteristics that require special machines to rectify, this problem has been satisfactorily solved with comparatively simple equipment. Even where it is desirable to transmit power under ground at high voltages, the problem is being met in a fairly satisfactory manner. Cables are in operation on alternating-current voltages up to 66,000. Although the characteristics of these cables may not be all that is desirable, the future holds forth promising prospects for their improvement. So that here again developments would not seem to warrant the introduction of the complications of a direct-current machine. Although the transverter has been built of a size and operated at a voltage that would indicate that those who are developing it have confidence in its future, the size of the machine is far from being one that would fit into our modern transmission system. Nevertheless, the daring and initiative of the manufacturers must be admired.—"Power," August 12, 1924.

PROPELLER DESIGN.

The Tendency of Airship Propeller Design to Follow that of the Marine Propeller.

Very interesting cases not infrequently arise in which technical history has an extraordinary knack of repeating itself. That of the evolution of the marine screw propeller is well known. At first it was geared up from the engine; then it was direct driven; then, directly driven by the steam turbine, it ran at unduly high revolutions; today it is geared down; but even now the tendency with gearing is to adopt somewhat smaller propellers run at higher revolutions than would be the case with the direct engine drive. It is somewhat remarkable that, although the media in which they work are so entirely different, the general tendency of air-screw design is following somewhat closely that of the marine propeller, and smaller propellers of relatively high tip speed are now being favored rather than those of lower revolutions and greater diameter, and which, in many cases, have been geared down from a very high-speed petrol engine. Some time ago we drew attention to the fact that the largest screw propellers fitted to ocean-going vessels were by no means those of modern design. Quite the largest of which any record is available was that of the *Great Eastern*, which was 25 feet in diameter by 44 feet pitch, but many of the large single-screw ships of the 'eighties had four-bladed propellers over 23 feet in diameter and 35 feet pitch, with large area values, and which, had they been driven, would undoubtedly have shown very large power absorption coefficients. The modern practice even in vessels of big power propelled by reciprocating engines is in favor of restricted diameter and higher revolutions.

With the increase in power of aircraft engines it is becoming expedient in many cases to employ air screws with relatively high tip speeds. There is, however, a limit in tip speed beyond which it would be very inefficient to run an air screw. Nothing is known as to the performance of air screws with tip speeds greater than the velocity of sound, but at tip speeds approaching that value the usual air flow breaks down completely. Experiment has demonstrated that as the tip speed increases to this critical value the slip stream rapidly diminishes and ultimately vanishes when the velocity of sound is reached. Air then appears to be sucked in from both sides of the

The following table gives a few particulars of the air screws fitted to recent rigid airships:

Airship	*R-26*	*R-29*	*R-32*	*R-33*	*R-38*	*LZ-126*
Length, feet	535	539	615	644	694	656
Total weight, tons	28.6	30	47	59.5	84	82
Engines	3 × 250 1 × 275	3 × 275 ...	5 × 250 ...	5 × 350 ...	6 × 350 ...	5 × 400 H.P. ...
Revolutions	1600	1800	1600	1900	1800	1400
Propellers, diameter feet	2 × 10 (in forward car driven by one engine)	2 × 10 (in forward and aft cars)	5 × 17 (one in each car)	1 × 17.4 (in each wing car and forward car)	2 × 17.5 (in two forward cars)	12 feet 1½ inch
	2 × 11.08 (in midships car each driven by one engine)	1 × 13.5 (in midships car)	...	1 × 19.5 (driven by two engines in after car)	4 × 17.3 (in four after cars)	...
	1 × 13.5 (in after car)	...	...	...	...	...
Propeller, R. P. M.	820 forward 1000 amidship 1024 aft	920 forward and aft 1150 amidship	620	420	540	1400
Tip speeds, feet per minute	25,750 34,800 43,600	28,900 48,800 ...	33,150	26,900 30,100 ...	29,750 29,400 ...	53,500 ... ...
Tip speeds, feet per second	428 580 726	482 813 ...	552	449 502 ...	496 490 ...	892 ... ...
Speed of ships, feet per second	70.0	84.5	102.6	88	103.6	111 estimat'd
M. P. H.	47.75	57.6	70	60	70.6	76 estimat'd

disc and exhausted at or close behind the periphery, under such conditions the thrust becomes very small and may vanish. The power absorbed on the other hand increases continuously with the cube of the revolutions. It will probably come as a surprise to most naval architects and marine engineers to realize precisely what tip speeds are actually involved in air-screw work. The highest values attained in large direct-driven marine propellers have touched 17,000 feet per minute or about 284 feet per second in the *M* class destroyers, and even under such circumstances have shown quite high efficiencies. The remarkable fact about air screws, however, is that they attain, and even exceed, peripheral velocities that would be regarded as extremely high if employed as blade speeds for steam turbines of disc construction.

The ship speeds for the British *R* class are those observed on official trials; the Zeppelin *L.Z.126*, which is destined for the United States, is not quite complete. It will be seen from the table that the tip speeds involved are very high. Even so they all, with the exception of the German vessel, compare favorably with current aeroplane practice. Among many other experiments which the Aeronautical Research Committee have undertaken in recent years was one which had as its motive the determination of the relative importance of the causes of noise in aeroplanes. With this object a long series of tests were made with a variety of machines of different size and power with the idea of attempting to separate the extent of the sound produced into the various factors of exhaust noises, radiator whistling, machine resistance sounds and propeller roar. The basic intention in conducting these tests was to discover, if possible, some means of eliminating the warning noise made by military aeroplanes, which is really the most attracting form of guidance to their presence at a distance. These experiments were not entirely conclusive, and certainly did not attribute the major share to the action of the propeller even though tip speeds of 600 to 660 feet per second were involved in the case of the D.H. machines with propellers exceeding 9 feet in diameter running at 1200-1300 revolutions per minute. This speed is fully 10 per cent higher than is now usual in the case of the low-pressure blading of high-power fast-running Parsons type land turbines, and when it is considered that the airship and aeroplane screws attain these tremendous velocities, no small degree of praise is due to the constructors who are working with such materials as walnut or mahogany and not with metal.

There are many other aspects of airship screw design which bear a remarkable resemblance to marine conditions. The fore and aft distance separating the cars containing engines and screws may be as much as 180 feet, or more than double the distance which separates the inner and outer pairs of propellers in the *Mauretania* or *Majestic*, yet the slip stream effect of the forward on the after sets of screws is very marked. The positions, moreover, in which the air screws should be placed to the best advantage is still very uncertain, and while, from the point of view of efficiency, a very high standard of actual performance has been obtained, all the features corresponding to thrust deduction and mutual interference which multiple marine propellers exert upon one another is being felt to a very considerable extent in airship work. Naval architects will require but little consideration to appreciate the innumerable problems in this direction which the designers of lighter-than-air craft are inevitably bound to encounter.—"Shipbuilding and Shipping Record," August 14, 1924.

DIESEL ENGINE COOLING.

For the metals in common use in marine Diesel engines today a satisfactory system of cooling is essential for successful operation. For sea service, what appears at first sight to be the most economical cooling

medium, as it is the most convenient, is ordinary sea water; but one serious objection to the use of sea water is to be found in the fact that when entering the rivers leading to numerous ports in the world, muddy water, charged with heavy refuse, finds its way into the restricted annular cooling spaces in cylinder jackets, covers and pistons, and the satisfactory cleaning of these confined spaces is often a practical impossibility. An alternative method of piston cooling is to use oil, but the high piston temperature which in the case of sea-water cooling results in depositing scale and other solid refuse is found to cause an equally objectionable deposit of carbon. Unless the oil-cooling supply is frequently purified or renewed, the oil soon reaches a state of plastic consistency which renders its further use impracticable. The frequent renewal of cooling oil is naturally very expensive, and as a result of this and of the drawbacks of salt-water cooling already described, many owners have been forced from their experience in operating motor tonnage to adopt the principle of fresh-water cooling. The latter system of cooling is not without its disadvantages, as the auxiliary machinery involves a greater initial outlay. With sea-water cooling one circulating pump is adequate, but with fresh-water cooling an additional cooler must be provided for cooling the circulating fresh water, and this involves also an additional fresh-water circulating pump. Experienced owners, however, regard such increased initial costs as being amply compensated by the freedom from deposits and the choking up of annular cooling spaces, with the result that specifications covering new constructions are increasingly calling for fresh-water cooling in main and auxiliary units.—"Shipbuilding and Shipping Record," September 4, 1924.

THE DIESEL ENGINE.*

By Dr. Adolph Nägel.

The Diesel engine has been developed along four principal lines in Germany since the war; first, the main type of the engine has been freed from certain handicaps caused by too close an adherence to the original model; second, the two-stroke cycle is being used for large engines; third, there is an increasing tendency to inject and atomize the fuel without the aid of compressed air; and fourth, much progress has been made in the use of fuel of low ignitability.

It has to be admitted that the Diesel engine without compressor became recognized in Germany at a comparatively late date, and that the leading firms, with very few exceptions, did not set great store by the airless injection of fuel. It was only when economic conditions made it imperative to lower the cost of construction that the interest in the simplified Diesel engine made itself felt. Today it can be stated that the economic pressure which makes solid injection necessary is felt by the German Diesel engineers as a far-reaching impetus for renewed activity.

Before the war many attempts had been made to separate part of the combustion air which had been compressed in the combustion chamber of the working cylinder, and to compress it still further in order to use it for atomizing. All these attempts failed because the different parts used in the new constructions afforded no safety in working. So long as this method is not revived in one form or another we have to distinguish between two types of Diesel engines only: the antechamber engine and the jet-spray engine.

With the former kind, the antechamber engine, the atomizing of the fuel is produced by an additional pressure of air which originates by means of an auxiliary explosion within the antechamber. Between the antechamber

* Paper, abridged, contributed to Section E of the World Power Conference on July 8, 1924.

and the working cylinder a nozzle or a system of nozzles is placed in order to convert the pressure of the antechamber gases into velocity and to utilize it for atomizing. The nozzle is the characteristic feature of the Diesel engines in contradistinction to the ignition head engines, which work with low compression and whose antechamber is not separated from the combustion chamber by a nozzle.

Whereas with the ignition head engine the spontaneous ignition is caused solely by the high temperature of those parts of the surface of the ignition head which are hit by the jet of fluid, the antechamber Diesel engine, in order to bring about spontaneous ignition, utilizes also the increase of the temperature of the combustion air as originated by compression in addition to the temperature of the inner surface of the antechamber.

With the jet-spraying engine one or several very thin jets of fuel enter the combustion chamber direct under the high pressure of several hundred atmospheres. The atomizing of the fluid is caused by the dissipation of these jets, which is brought about through the influence of the friction of the jet on the compressed charge of air. As soon as this principle is realized it becomes evident that we must allow the jet to travel within the combustion chamber a distance sufficient for its dissipation. If it should strike the wall too soon the dissipation would not be complete and the quality of combustion would be appreciably lowered. This is the reason why a complete success was scored when several years before the war horizontal Diesel engines were built with solid injection.

The peculiar construction of the combustion chamber of this horizontal engine afforded sufficient travelling space for the jet to dissipate before it could hit the surface of the piston as a solid jet. Conditions differed, however, when an attempt was made to adopt this mode of dissipating the jet for the usual type of vertical Diesel engines, where the jet of the fluid, entering axially and travelling only a few centimeters, hit the surface of the piston directly. The failure resulting from this impeded for a long time the development of the solid injection engine of the vertical type, until it was recognized that it was the short travel of the solid jet which prevented its even dissipation, a shortcoming which has been overcome by a hemispherical depression of the piston surface.

Fig. 1 shows the needle-tube arrangement adopted in one design. This opens automatically under the pressure of the fuel. Under it the head of the piston on the upper dead center is to be seen. All tests with several engines of this type resulted in records of efficiency. The latest tests were made by Professor W. Maier, Stuttgart, who obtained the values registered in Fig. 2. If these are reduced to a net calorific power of 10,000 calories per 1 kg., and if a normal load of 360 E.H.P. is assumed, we obtain the following values of fuel consumption per E.H.P. per hour:

¼	load, grams	224
½	load, grams	177
¾	load, grams	168
1/1	load, grams	167
and		
5/4	load, grams	177

This exceedingly small consumption of fuel is remarkable. The minimum of 166 grams was attained with a load of 320 E.H.P., which corresponds to a thermal efficiency of 38.3 per cent. It must also be emphasized that the fuel consumption is in a large measure independent of the load; within the wide range from half-load to 25 per cent overload the minimum fuel consumption is exceeded only by 6.6 per cent. A careful analysis of the underlying causes has shown that only about 50 per cent of this favorable result is due to the absence of the compressor work. A large part of it is

to be attributed to the hemispherical, practically uniform combustion chamber, whose cooling surface is comparatively small, as well as to the high mechanical efficiency, amounting to 85 per cent. The engine stands an overload of nearly 50 per cent. Professor Maier was able to make a satisfactory test with a fuel consumption of 188 grams at a mean effective piston pressure of 7.83 atmospheres. As the normal load of 360 E.H.P. corresponds to a mean effective pressure of 5.5 atmospheres, the overload is

$$\frac{7.83-5.5}{5.5}=42 \text{ per cent.}$$

Experience seems to point out that the solid injection engine of this type very favorably influences the wear of the piston, whose upper surface is very effectively cooled by the contact with the spray of the fuel. The heat thus drawn from the piston surface is used for vaporization and for cracking the small fuel drops. It is believed that for this reason a separate cooling of the piston can be dispensed with for larger diameters than is the case when the spraying is effected by compressed air.

Very similar devices have been successfully used by other firms. The fuel pump in one type of four-stroke cycle engine is worked by a cam projecting from the crankshaft. In case of a two cylinder engine the fuel is admitted alternately into either cylinder by means of a distributor which is connected with the gear-shaft; with single-cylinder engines on every second stroke of the pump the fuel is by-passed to the fuel tank. Fig. 3 shows the injection nozzle. The needle, which opens automatically under the pressure of the fuel, is provided with a lift-stop in shape of an inverted mushroom valve. If the needle is lifted this valve presses tightly against the lower surface of the upper bush of the needle guide. In this way the packing of the needle can be dispensed with, and a weak spring will prove sufficient for loading the needle. The nozzle proper has five pairs of holes which are equally distributed on the circumference. Each pair consists of a horizontal and an oblique bore, each 0.3 millimeter in diameter. Both bores of each pair meet at the point of discharge; thus the two jets combine and the atomizing of the jets occurs. This engine has yielded a fuel consumption as low as 163 grams per effective horsepower-hour.

A large number of engineering works are also interested in the antechamber Diesel engine. This type does not yield the exceedingly low fuel consumption which is the feature of the most economical types of the jet-spray engines. But it is also very advantageous as it requires less precision in manufacture, dispensing as it does with the two most sensitive parts of most of the jet-spraying engines, *viz.*, the injection nozzle and the needle mechanism. According to our latest experience it is a matter of doubt whether preference is to be given to the antechamber engine or to the jet-spray engine if we sum up the advantages of either system, such as low prime cost, safety in working, easy operation by the factory hands, and, last but not least, low fuel consumption. I discussed the chief types of antechamber engines which are now being built in Germany in a lecture delivered to the Verein Deutscher Ingenieure in Berlin last year. The feature of the Steinbecker antechamber engine is the introduction of the fuel into a nozzle zone which is placed between the antechamber and the working cylinder. Through the current of air which flows, during compression, from the working cylinder to the antechamber, the small particles of fuel which are first introduced into this zone are carried into the antechamber, whose walls, brought to blood-red heat, cause the spontaneous ignition of this very small amount of fuel. In consequence of the combustion pressure in the antechamber, which suddenly rises to 80 atmospheres, the direction of the current within the nozzle is at once reversed, and this new current causes the bulk of the fluid which now follows to be injected into the cylinder and atomized with the aid of the gas pressure in the antechamber.

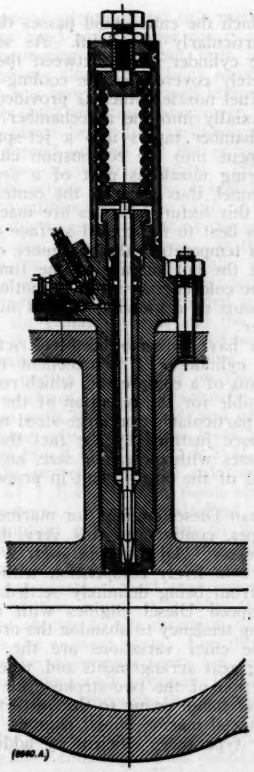


FIG. 1.—Needle Tube Injection for Vertical Diesel Engine.

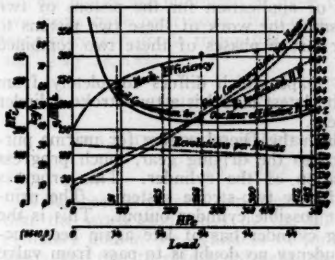


FIG. 2.—Results of Tests made by Professor W. Maier, Stuttgart.

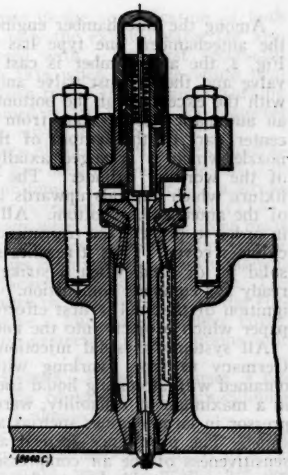


FIG. 3.—Needle Tube and Spraying Nozzle for Four-Cycle Engine.

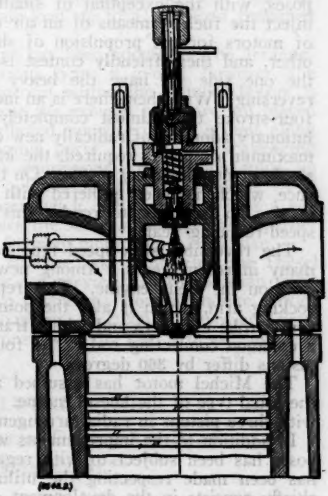


FIG. 4.—Antechamber Diesel Engine.

Among the antechamber engines, in which the entire fluid passes through the antechamber, one type has been particularly successful. As seen in Fig. 4, the antechamber is cast into the cylinder cover between the inlet valve and the exhaust valve and completely covered by the cooling water with the exception of the bottom. The fuel nozzle, which is provided with an automatic needle, runs from above axially into the antechamber. The center part of the bottom of the antechamber tapers into a jet-spraying nozzle which is arranged axially and opens into the combustion chamber of the working cylinder. The jet-spraying nozzle is part of a separate fixture which expands upwards into a funnel, thus forming the center part of the antechamber bottom. All around this fixture grooves are machined, in order to prevent the transmission of its heat to the cooled surface of the cylinder cover. Thus it reaches so high a temperature that the more or less solid jet of fluid, when it strikes against the hot surface of the funnel is ready for complete combustion. When the cold engine is put in motion, the ignition of the fuel is first effected by means of a glowing fuse of nitrated paper which projects into the antechamber.

All systems of solid injection engines have hitherto been restricted in Germany to types working with small cylinders. The excellent results obtained with atomizing liquid fuel by means of a compressor, which resulted in a maximum of reliability, were responsible for the retention of the compressor in large engines such as are used particularly for large-sized motorships. The preference seemed all the more justified by the fact that the sensitiveness of the air compressor decreases with increasing size, and that the compressor adds less to the prime cost of the entire plant in proportion as the size of the engine increases.

All these reasons explain why the German Diesel engines for marine purposes, with the exception of smaller types, continue to this very day to inject the fuel by means of an air compressor. All these different systems of motors for the propulsion of ships are in lively competition with each other, and their friendly contest is far from being definitely settled. On the one side we have the heavy slow-speed Diesel engines with direct reversing. With them there is an increasing tendency to abandon the original four-stroke type almost completely. The chief variations are the revolutionary adoption of radically new driving gear arrangements and, wherever maximum output is required, the introduction of the two-stroke cycle with single or double-acting piston. On the other hand, owing to the vast experience, which has been gathered with high-speed marine Diesel engines, there is also an endeavor to render this latter type more efficient by adding a speed-reducing gear.

The reversible slow-speed type has been considerably developed in Germany in recent years. Among new driving gear devices we have first to mention the Benz engine. This replaces the cross-head guide by a strong rocking arm, which is also the point of application for the pistons of two four-stroke cylinders and which transmits the work of these two pistons to a common connecting rod. The four stroke phases of these two combined pistons differ by 360 degrees.

The Michel motor has assumed a shape which differs completely from the usual type of the Diesel engine. Its latest model is a two-stroke cylinder with three pistons in radial arrangement.

In addition to the improvements which the Diesel engine for marine purposes has been subject to with regard to the driving gear, much progress has been made respecting the utilization of the cylinder. This progress chiefly consists in the development of new two-stroke systems. The principal endeavor is to obtain the highest possible cylinder output. This is the reason why the double-acting working cylinder has of late again been successfully resorted to. The modern tendency no doubt is to pass from valve scavenging to port scavenging which is controlled by the working piston.

The special advantage of this method consists in the simplification of both cylinder head and cylinder cover, since not only the exhaust valve but also the scavenging valves can thus be dispensed with. The first successful experiments along these lines were made in 1914 when a single-acting trial engine was tested of the well-known two-stroke type. In the single working cylinder 1992 effective horsepower was obtained at 150 R.P.M., the diameter of the piston being 1000 millimeters and the stroke 1100 millimeters.

Of special interest are the excellent results obtained with a new double-acting two-stroke trial engine. This engine has been thoroughly tested with a double-acting, two-stroke cylinder of 800 millimeters diameter and a stroke of 1050 millimeters. The ratio of the stroke is comparatively small (1:3); this is due to certain parts of the engine which were on hand and were made use of for the trial engine. With ordinary engines a larger ratio will be chosen for the stroke. Particular attention has been paid to an advantageous circulation of the cooling water within the jacket of the cylinder liner. The upper cylinder cover is provided with a fixture in which are placed the fuel valve, the exhaust valve, and the safety valve. The lower cylinder cover, which is provided with a cooled wrought-iron protection plate, contains four fuel valves and one safety valve.

This engine was started on December 18, 1923. From the outset it has been working very satisfactorily, so that the construction of six-cylinder engines with 10,000 E.H.P. and 94 R.P.M. is planned.

Taking into account the work of the scavenging pump, the trial engine yielded 984 E.H.P. at 88.7 R.P.M. The fuel consumption, reduced to a net calorific power of 10,000 calories per kilogram, was 187 grams per one hour of 1 E.H.P. The mean effective piston pressure was as high as 5 atmospheres. With a half-load, *i.e.*, 515 E.H.P. at 89 R.P.M., the fuel consumption only went up to 200 grams. Calculated per one hour of 1 I.H.P., the fuel consumption for a half-load up to a full load remained constant at 150 grams.

Quite recently a port scavenging engine has been adopted. The arrangement of this engine resembles the previous type; it avoids, however, the sudden change in the direction of the current within the cylinder before it enters the exhaust ports. For this purpose the exhaust ports are not placed above the scavenging ports, but are distributed sidewise in two groups along the cylinder wall opposite the scavenging ports. Thus the scavenging current flows through the cylinder in a continuous circle which splits up into two currents in order to leave the cylinder at the exhaust ports. For smaller marine Diesel engines, up to about 400 E.H.P., one firm uses a sun and planet coupling. This coupling offers the same advantages for smaller marine engines as the Vulcan gear does for ship motors of a large size.

A final word may be said about the similar task of utilizing the Diesel engine for locomotive and freight motor-car service. Many German engineering works are interested in these two problems. For locomotive service two firms have been using the Lentz coupling. A 1200 horsepower Diesel locomotive for goods trains has been constructed by one firm.—"Engineering," July 18, 1924.

DIESEL-ENGINE LUBRICATION.

The Employment of One Grade of Oil for All Purposes.

By Chief Engineer Ernst.*

It has been a general experience in internal-combustion-engine lubrication that the work performed by the oil in different parts—*e.g.*, in the main cylinders and in the cylinders of air-injection compressors—is of different kinds, and that therefore different oils should be used. But, apart from this,

* Extract from a paper read at Hamburg.

the manner in which the oil is led to the different parts of the engine is of importance, and if the system of circulation is constructed accordingly one grade of oil may be used for all parts, and good results have been attained in this way.

The systems of oil circulating which are generally applied to Diesel engines are different, according to the size, type and age. Horizontal engines of small size are provided with normal circulation without pressure; with larger engines of the vertical type forced lubrication is used. With the older systems the lubricating oil from the crank bearings and from the cross-head is sprayed into the open lower part of the cylinders, thus augmenting the lubrication of the cylinders beyond the intended amount. This leads to combustion of the lubricating oil in the cylinder. The combustion of lubricating oil always occurs if the lubrication is excessive; it then gains access more easily to the heated parts of the combustion chamber. But even in these regions the temperature is not sufficient to effect complete combustion of the lubricating oils, and the incomplete combustion naturally results in carbonization.

Tar-oil tends more to incomplete combustion than mineral oil, and to the formation of soot in the cylinder, by which the lubricating oil is polluted when the excessive lubricating oil leaks down into the crankcase. This naturally leads to congestion of the circulation.

SEPARATION OF CRANKCASE FROM CYLINDER.

The construction of the engines should, therefore, be such (and this is done with all modern types) that the crankcase is separated from the cylinder, so that no oil can be sprayed into the cylinders, and the amount of lubricating oil delivered to the cylinders by regular lubrication can be controlled. It has been found by experience that it should be as small as possible, in order to avoid combustion of the oil. If the crankcase is separated the lubricating oil coming from the cylinder may be kept more clean, as shown above, but also the oil in the general circulation is kept free from impurities coming from the cylinders and from oil or water from the piston-cooling system.

As regards the cooling of the pistons, it may be effected by the general circulating lubricating system if the oil is fit for the purpose. The construction of the cooling arrangement must be such that those parts of the piston head exposed to the heat of the combustion flame from the injection valve are well cooled by the oil. In small pistons no special measures are necessary for this purpose, but with larger pistons pressure must be applied, and the oil must be led in spirals that are formed by the lower part of the piston head, and by the head of the cooling pipes. If there are two or more injection valves, special care must be taken to lead the cooling oil to the hottest parts and to make it flow through the hollow edges. If this be done, the formation of scale deposits can be avoided; this is important, for such deposits form a severe danger. They hinder the transmission of heat and the cooling of the piston head, and they may pass into the general circulating lubricating system, which may be congested by them. With very large pistons the cooling effect of oil is insufficient, as its thermal heat is only 0.45 that of water. Here water cooling is better, but it must be prevented from getting into the lubricating oil, especially if sea-water is used, as it tends to form a soapy emulsion with some kinds of oil. But with most engines of older types there is no separation of the crankcase, and, as leakages from the piston cooling are unavoidable, it must be the aim of oil suppliers to produce lubricating oil from which the water may easily be separated.

TEMPERATURE FOR SEPARATION.

When considering the stresses to which the lubricating oil is exposed during its passage through the engine, it must be observed that there is not only the mechanical stress by pressure in the bearings and the gears, but also a very high thermal stress, and the contamination by coke, soot, sea-water and other impurities. Cleaning the oil by mechanical strainers in the course of the circulation has been found to be insufficient. Also the centrifugal separators generally used, in which the oil is heated up to about 160 degrees F., are often insufficient for the purpose. Steam-heated separators with a temperature of 210 degrees F. are much better, but in some cases it has been found necessary to apply pressure-centrifugal separators with a temperature of about 270 degrees F. There have been observed cases in which the lubricating oil had to circulate 15 times per hour through the Diesel engines. It should be a general rule to have larger quantities of oil in the circulating system than are usually employed; then the oil is not so quickly vitiated, so that cleaning is easier and it may pass to a settling tank where water may be drawn off.

CASES OF LUBRICATING OIL TROUBLES.

Two interesting cases were noted in which disturbances of the lubrication caused difficulties. The first was with a vertical Diesel engine, where the pistons were cooled by sea-water. It had run well for some months with an oil that served for lubrication to all parts; then the circulation to a connecting-rod bearing ceased owing to congestion by an oily mud. It was found that oil had dropped from the cylinder regularly into the crankpit, and that the oil had been vitiated by soot, which was deposited at the end of the lubrication piping near the bearing of the connecting rod. The mud proved to consist of 4 per cent of carbon, 6 per cent of water and 90 per cent of oil. Even this small percentage of carbon was enough to thicken the oil too much. The oil, after being properly cleansed, proved to be of good quality. There had been normal oil strainers on the ship for use in general service, and when the vessel stopped the oil was collected in a settling tank, the mud from the lower part was drawn off and the other oil cleansed by the centrifugal separator, heated by warm water; but these cleaning arrangements proved to be insufficient.

The second difficulty was experienced with a high-speed-geared engine, where a single kind of oil was used for the lubricating of the bearings and the cooling of the engine pistons. The circulating oil of this engine deteriorated after running for some time, although it was regularly strained and separated. It was so much thickened that it could not be worked by the normal pumps, and an auxiliary pump was used in addition. Investigation of the oil showed that there was 14 per cent of free water in it. Further, 65 per cent of emulsified water was found, 20 per cent of oil and only about ¾ per cent of carbon. This trifling amount contained in very fine parts in the mixture of oil and water was no doubt the cause of the thickening of the oil. There were no pieces of hard coke from the piston cooling in the oil. The mixture could only be separated in pressure-centrifugal separators heated to 270 degrees F. The oil thus gained proved to be absolutely pure, and was equal to fresh oil apart from a change of color. The analysis of the oils which were separated from the mud of the two above-mentioned cases in comparison with the analysis of the fresh oil showed that its quality is but very little altered, as may be seen by the accompanying table:

These figures show that even very hard service does not materially alter the quality of good lubricating oil.

	Fresh Oil.	Oil separated from mud Example No. 1.	Oil separated from mixture Example No. 2.
Specific gravity at 59 degrees F.	0.9058	0.9069	0.9093
Viscosity in Engler degrees at 122 degrees F.	5.2	5.36	5.19
Flash point, degrees F.	444	420	438
Acid number.	0	0.484	0.218

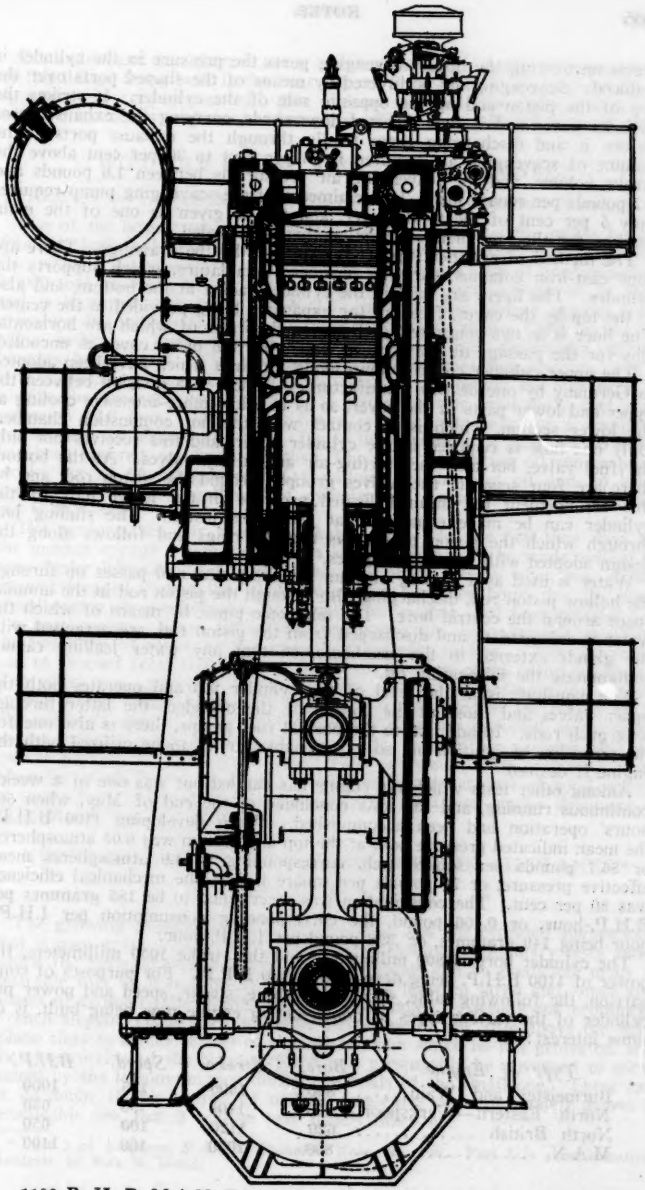
Altogether, the lubrication of Diesel engines may be regularly effected with advantage by some different kind of oil for the various parts of the engine. It may be mentioned that, contrary to early opinion, oils of low viscosity may be used for the lubrication of cylinders of injection-air compressors. If simplicity of operation is required, and if the construction of the engines is executed accordingly, there is no difficulty in applying a single oil for all purposes—*i.e.*, lubrication of the cylinders of the main engines and of the compressors, as well as the bearings and the gearing, applying it also for cooling the pistons and filling the impellers of hydraulic couplings. It would not be of value to give a general scheme of analyzing data required from oils for special purposes, as they are too different. But experiences with Diesel engines and consideration of special features may lead to results of co-operative work of the engine builder, the user and the oil producer, which will be useful for the solution of the questions still existing in connection with Diesel oil-engine lubrication.—"The Motor Ship," August, 1924.

THE M.A.N. DOUBLE-ACTING ENGINE.

A 1100 B.H.P. Single-cylinder Two-stroke Type with Port Scavenging.

In November, 1923, we gave a few details of a new type of double-acting two-stroke engine being developed by the Maschinenfabrik Augsburg Nuremberg, with a sectional drawing of the cylinder. Since that time work has considerably advanced, and a long series of trials has recently been concluded upon a 1100 B.H.P. experimental cylinder at the Augsburg Works. An interesting article concerning this new engine is published in our contemporary "Werft Rederei Hafen," by Dr. Riehm, who is the chief engineer of the M.A.N. Augsburg Works. From the sectional and outline illustrations of this motor, for which we are indebted to our contemporary, it will be seen that in the design of cylinder liner and mantle a somewhat similar arrangement has been employed as in the cylinder of the 12,000 B.H.P. six-cylinder double-acting two-stroke engine which was built by the M.A.N. during the war and destroyed. Full details were published of this a short time ago in "The Motor Ship."

A very important modification has, however, been made, for port scavenging is now utilized instead of the employment of large, cumbersome and complicated valves. A great deal of experimental work has been carried out with the object of attaining the maximum efficiency in scavenging. It is somewhat surprising to find that the scavenging ports, also those for the discharge of the exhaust gases, occupy only a portion of the periphery and are, moreover, arranged on the same side of the cylinder. There are two rows of each, the exhaust ports in the upper part being above the scavenging ports and in the lower part below them. As the piston descends it first uncovers the upper series of exhaust ports, allows the exhaust pressure to fall and part of the exhaust gases to be discharged. By the time the piston



1100 B. H. P. M.A.N. Double-acting Two-cycle Diesel Engine.

starts uncovering the upper scavenging ports the pressure in the cylinder is reduced. Scavenging air is directed by means of the shaped ports over the top of the piston and up the opposite side of the cylinder. It strikes the cylinder cover and is then directed downwards, carrying the exhaust gases before it and discharging them finally through the exhaust ports. The volume of scavenging air required is 25 per cent to 30 per cent above the stroke volume, and the scavenging air pressure is between 1.6 pounds and 1.7 pounds per square inch. It is claimed that the scavenging pump requires only 5 per cent of the total power, and this is given as one of the main reasons for the low fuel consumption that is recorded.

The method of cylinder support is explained by the drawings. There are four cast-iron columns, carrying a square entablature, which supports the cylinder. The liners are held in the cylinder jacket at the bottom, and also at the top by the cover, freedom for expansion being provided in the center. The liner is in two concentric sections, on the inner of which are horizontal ribs for the passage of the cooling water, but the outer cover is uncooled.

The upper cylinder cover follows along the lines which have been adopted in Germany by one or two manufacturers. There is a division between the upper and lower parts of the cover, so as to allow more intensive cooling at the lower section, which is in contact with the hot combustion chamber. Only one hole is required in the cylinder cover, and this receives not only the fuel valve, but also the starting-air and safety valves. At the bottom there are four separate fuel valves grouped around the piston rod, and by this arrangement the mean indicated pressure in the lower half of the cylinder can be made equal to that above the piston. The stuffing box through which the piston passes has metallic rings and follows along the design adopted with large gas engines.

Water is used as a cooling medium for the piston and passes up through the hollow piston rod, discharging also through the piston rod in the annular space around the central bore. The telescopic pipes, by means of which the water is delivered to and discharged from the piston rod, are arranged with the glands external to the crankcase, so that any water leaking cannot contaminate the lubricating oil.

The camshaft is at the level of the cylinder top and operates both the upper valves and those at the bottom of the cylinder—the latter through long push rods. In addition to the normal fuel pumps, there is also one for the provision of ignition oil, so as to enable tar oil to be utilized with the engine if desired.

Among other tests which the engine has carried out was one of a week's continuous running, and the tests concluded at the end of May, when 500 hours' operation had been accomplished. When developing 1100 B.H.P., the mean indicated pressure both at the top and bottom was 6.05 atmospheres, or 86.7 pounds per square inch, corresponding to 4.9 atmospheres mean effective pressure, or 70 pounds per square inch. The mechanical efficiency was 80 per cent. The consumption was ascertained to be 185 grammes per B.H.P.-hour, or 0.406 pound, the corresponding consumption per I.H.P.-hour being 149 grammes, or .328 pound per I.H.P.-hour.

The cylinder bore is 800 millimeters and the stroke 1050 millimeters, the power of 1100 B.H.P. being developed at 100 R.P.M. For purposes of comparison, the following table, showing the bore, stroke, speed and power per cylinder of the various types of double-acting engine now being built, is of some interest:

Type of Engine.	*Bore.*	*Stroke.*	*Speed.*	*B.H.P.*
Burmeister and Wain.....	840	1500	125	1000
North Eastern—Werkspoor.	800	1400	95	650
North British	620	1120	100	650
M.A.N.	800	1050	100	1100

We understand that there is every likelihood of large double-acting engines of this design being built in the near future, to be fitted in a passenger liner, operating in conjunction with the Vulcan hydraulic mechanical gear, to which reference is made on another page.—"The Motor Ship," August, 1924.

TWO-CYCLE DIESEL ENGINE PERFORMANCE.

Results of Eleven Months' Service with a 14,000-ton Tanker.

One of the largest two-cycle-engined ships in service is the motor tanker *Phœbus*, built last year to the order of the German American Petroleum Co., which is allied with the Standard Oil Co. It is a ship of approximately 14,000 tons deadweight, being 498.9 feet in length, with a beam of 64 feet, and is equipped with two 1600 B.H.P. four-cylinder Sulzer engines with cylinders 680 millimeters bore and 1100 millimeters stroke (26¾ inches by 43⅜ inches). Turbo-scavenging is employed, two electrically driven blowers being fitted. The generating plant comprises two engines of 270 B.H.P. each, running at 180 R.P.M. During the period she has been in service the average speed has been 11 knots and the total daily fuel consumption 14.2 tons. The lowest figure is 13.5 tons daily for a complete voyage and the highest 14.5 tons. Up to the beginning of July, 66,000 miles had been covered, *i.e.*, in 11 months. The daily lubricating oil consumption works out at 50 kilogrammes (11 gallons) of bearing oil and 30 kilogrammes (about 6½ gallons) of cylinder oil, equivalent to 1.1 grammes per B.H.P.-hour.

The performance has been remarkably good. In the first few days of the maiden voyage a few stops for a short time had to be made, owing to some leaky lubricating-oil pipes. The only other periods when any of the engines stopped during the whole time were for four hours for the starboard engine on one occasion, 15 minutes on another, and four minutes for the port engine. Some interesting experience has been gained regarding the suitability of sea-water for piston and cylinder cooling. Once the vessel had to proceed from Galveston to Baytown, the time taken being six hours. The water was extremely muddy and the vessel went aground for a time. It was found, however, that no solid matter deposited in the piping or in the cylinders and pistons. The sea-water filter is provided with plates having holes 5 millimeters in diameter. It is therefore claimed that no special precautions have to be taken when sea-water cooling is used, even when the vessel has to proceed in waters with a good deal of solid matter in suspension. This is a view not taken by all Diesel-engine designers.—"The Motor Ship," August, 1924.

NOTES ON GEOMETRICAL SIMILARITY IN AIRPLANES.*

By Edward P. Warner.

The growing popularity of the light 'plane, and the repeated suggestions that it may prove a satisfactory vehicle for making preliminary tests from which the performance of much larger airplanes can be predicted, make it desirable that an investigation of the relation between large and small airplanes of geometrically similar form be undertaken. Already one prominent French airplane constructor has built a machine of but little above the light 'plane class as a scale model of a giant airplane which he has projected, and the construction of the large airplane will presumably be governed to some extent by the lessons learned during the trials of the small one. There can be no doubt, if this project of making man-carrying scale models proves a practicable one, that it will be very widely taken up.

* Part 2 of American N.A.C.A. Technical Note, No. 190. Part 1 is a mathematical Analysis, by Max M. Munk.

Instead of seeking to establish perfectly general relations between the performances of a large airplane and a small one, it seems desirable rather to determine the ratios which should exist between certain geometrical characteristics in order that the performances may stand in some particular desired relation. The elements of performance, including all the flying qualities of the airplane within that term, are measurable in terms of length, time and angle, as fundamental quantities, those quantities appearing either singly or in combination. Obviously, those elements of performance measured in terms of angle, such as the inclination of the climbing path, should be independent of the size of the airplane, while those measured in length, such as the minimum radius of turn, should be directly proportional to the linear dimension. The question of the variation of those elements into which time enters may be put aside for the moment, except for mention of the obvious fact that the linear velocities must be proportional to the product of angular velocities and linear dimension.

The various elements of performance and the characteristics of the airplane will now be taken up and investigated one by one. It would, of course, be possible to rely on the general theory of dimensions in deriving the desired relations, but conclusions can be extended over a somewhat broader range if each feature of performance is analyzed separately by methods simpler, and in some cases less rigorous, than the general theory.

MINIMUM RADIUS OF TURN.

Centripetal force in turning is proportional to the product of weight, angular velocity and linear velocity. If airplanes of different size are to turn at the same angle of bank and with the same control setting the ratio between centripetal force and weight must obviously be constant, and the product of angular and linear velocities must therefore be independent of linear dimension. The square of the speed must then be proportional to the radius of the turn for a given condition of flight. Since the radius is a distance, it should vary as the first power of a dimension of the airplane, if strict similarity of performance is maintained, and the speed must therefore be proportional to the square root of such a dimension of the airplane. It follows also that the angular velocity is inversely proportional to the square root of a linear dimension. Since the loading of the wings varies directly as the square of the speed for a given angle of attack, $\frac{W}{S}$ must be proportional to the first power of a dimension of the airplane, and the total weight must vary as the cube of such a dimension. For similarity of performance there is then the same rule of variation of weight as for strictly geometrically similar structures.

If this relation of weight and size be followed, not only the minimum radius of turn, but also the radius for any specified set of conditions will be proportional to the first power of a length in the airplane. For any fixed angle of attack and angle of bank the ratio of turning radius to wing span will be independent of size.

CONTROLLABILITY IN TURNING.

The maximum angular velocity of an airplane is sometimes fixed by the power of the control to overcome the damping of the rotation, rather than by a simple balancing of centripetal force and horizontal component of lift. Damping moments are proportional to the product of an angular velocity, a linear velocity, and the fourth power of a linear dimension, while control moments vary as the cube of a length and the square of a linear speed. In order that controlling power may enter in as a limitation in the same way for a whole series of geometrically similar airplanes, it is therefore sufficient that $l^4V^2=l^4V\omega$, or that $V=\omega l$. Obviously, any relation between speed and

size which will make radius of turn proportional to length of airplane will also satisfy this equation. The constant ratio between radius and length will therefore hold good, no matter what the factor principally limiting radius may be.

ANGULAR ACCELERATION.

The angular acceleration of an airplane for a given control setting is, of course, proportional to the ratio of controlling moment to moment of inertia. The first of these quantities varies as the cube of a linear dimension and the square of a speed, the second as the product of the weight and the square of a length, and the ratio is therefore proportional to $\frac{V^2 l}{W}$. If the relation, already derived, between l, V, and W be complied with, this varies inversely as a length. The time required to reach a specified velocity at constant angular acceleration would therefore be proportional to the linear dimension of the airplane, but, since the maximum angular velocity itself varies inversely as the square root of a length, the time required to reach the maximum is proportional only to $\sqrt{l}$. Distance being the product of time and speed, the distance covered in reaching maximum angular velocity or any particular fraction of the maximum must be directly proportional to l, and the angle through which the airplane turns from the beginning of a maneuver until a particular proportion of the maximum attainable angular velocity has been arrived at is quite independent of dimension. This, again, is as it should be.

DYNAMIC STABILITY.

While on the subject of control and maneuvering power, attention may be given also to the equations determining the amplitude and period of the oscillations of an airplane. The work need not be followed in detail, but if the variation of each of the resistance and rotary derivatives be examined separately, it is found that the coefficients in the familiar stability equation:

$$A\lambda^4+B\lambda^3+C\lambda^2+D\lambda+E=0$$

vary with l in a descending scale of powers of $l^{1/2}$, the first coefficient varying as l^2. This is obviously equivalent in effect on any solution of the equation to a variation of λ, and logarithmic determinant itself, in the ratio of the inverse square root of l. Since the amplitude of an oscillation at any time subsequent to its beginning is equal to $Ce^{\lambda t}$, the time required to damp or increase the amplitude of oscillation by a definite ratio must, obviously, be proportional to $\sqrt{l}$, if λ varies as the inverse square root. V and t then change with l in the same way, while the distance required to damp an oscillation by a specified amount or to complete one period of an oscillation is proportional to the product of V and t, or to a linear dimension of the airplane. Thus, once again, two lengths vary in the same ratio.

MINIMUM SPEED.

The minimum speed, being proportional to the square root of the wing loading, evidently varies as $\sqrt{l}$. The kinetic energy possessed by the airplane on coming in contact with the ground then varies as l^3, and, assuming the coefficient of friction the same in all cases, the landing run during which friction and air resistance dissipate this kinetic energy varies directly as a linear dimension of the airplane.

MAXIMUM SPEED.

Obviously, in order that performances may be comparable, geometrically similar airplanes should fly at maximum speed at the same angle of attack.

The maximum speeds must therefore vary in the same ratio as the minima. At a given angle of attack the power required for flight is proportional to the product of the area and the cube of the speed. If the speed varies as $\sqrt{l}$, and the propeller efficiency is constant, the power must then be proportional to $l^{3/2}$. To satisfy maximum speed requirements in a series of geometrically similar airplanes the engine power must therefore vary somewhat more rapidly than the weight.

PROPELLER EFFICIENCY.

In order that propeller efficiency may be constant, the slip function $\frac{V}{ND}$ must be held at a constant value. If V varies as $l^{1/2}$, N must therefore be inversely proportional to $\sqrt{l}$. This condition satisfies the equation of propeller power absorption also. Since the power consumed by a propeller at a constant value of the slip function is proportional to N^3D^5, it will vary as $l^{3/2}$, if N changes with size in the manner just stated.

SPEED AND ANGLE OF CLIMB.

The ratio between the powers required for flight at two particular angles of attack is obviously independent of airplane size. If the maximum speed corresponds to the same angle of attack in every case and the propeller characteristics are in accordance with the relation just derived, the percentage of reserve power at the angle of best climb will then be the same for a whole series of geometrically similar airplanes. Dividing reserve power by weight it appears that climbing speed varies as $l^{1/2}$, or, in the same ratio as speed of flight. The climbing angle is therefore the same in all cases.

LINEAR ACCELERATION.

The linear acceleration of an airplane in taking off is proportional to the ratio of thrust to weight, and that is obviously constant, if the prescribed relation between power and other characteristics of the airplane be preserved. The distance travelled in acquiring a given velocity is then proportional to the square of this velocity, and that, in turn, is proportional to l.

It has now been seen that all of the flying qualities of a small airplane can be made directly comparable with those of a large one, if a very simple relation between size, weight, power, and R.P.M. is maintained. That could have been predicted from the general theory of dimensions, following the line of Dr. Munk's work, and, indeed, the relations derived are identical with those given by Froude's law of comparison, and used for ships.* There are some points, however, at which similarity of performance breaks down. As pointed out by Dr. Munk, the condition of aerodynamic similitude, which would make the speed inversely proportional to a linear dimension, cannot be maintained, and the relation existing between flying characteristics in large and small sizes will also be modified by any structure in the atmosphere, a structure which will necessarily have linear dimensions of its own. Either periodic gusts or regions of turbulence will have effects depending largely on the size of the airplane which meets them. It is therefore somewhat unsafe to attempt to predict the behavior of a giant airplane in rough air from tests on a miniature prototype, but there need be no hesitancy about the application of data thus obtained on performance and on maneuverability under good conditions. The variation of the Reynolds number is unlikely to have any serious effect after values even as large as those for the smallest of light planes have been reached.

* "Speed and Power of Ships," by D. W. Taylor, p. 26.

STRUCTURAL RELATIONS.

The fact that it has been found necessary to vary the weight as the cube of a linear dimension suggests the possibility of building the structures in strict geometrical similarity, in order that the percentage of weight allotted to each part may remain the same in all cases. That would, indeed, be highly desirable for complete similarity of performance, as the radii of gyration are hardly likely to vary in the same manner as the overall dimensions, unless all the internal structure is kept of similar form as size is changed.

It is, of course, impossible to hold rigidly to similarity of structure. The thickness of fabric, for example, can hardly be decreased in proportion to the wing span, and the type of joint used in built-up members of large airplanes can hardly be duplicated in small ones. To a certain point, however, similarity can be maintained if it proves to be structurally safe to hold to it.

Considering first those members which are loaded directly in tension or compression, it is obvious that their strength is proportional to l^2. This is true even of long struts, since the ratio of l to k will be independent of the size of the airplane. The load carried by such members is proportional to the airplane weight, and the factor of safety in them therefore varies inversely as a linear dimension. Over the range of sizes now used, this is just about the desirable rate of variation, as it will be found that the load factors now specified for a high angle of attack are given approximately for all classes of military airplanes by the formula: $F=\frac{320}{b}$, where b is the wing span.

A similar relation holds true for beams. The bending moment varies as the weight of the airplane times the span, or as the fourth power of a linear dimension, while the section modulus is proportional to l^3. The factor of safety at a given load factor again changes inversely with l. When the beam is subject to buckling, however, the relation is no longer simple. The column effect is approximately allowed for by Perry's formula: $M^1=M\times\frac{P_e}{P_e-P}$, where M is the bending moment due to lateral load, M^1 the bending moment corrected for buckling, P the compressive load, and P_e the load which would produce failure by lateral collapse if there were no lateral load at all. P is proportional to l^2, P_e to l^3, and the corrected bending moment under unit load therefore changes with linear dimension in an irregular fashion. If, however, the load factor assumed to act itself varies as the inverse first power of l, P_e and P will change at the same rate and the column effect will remain always of the same relative importance.

It is also of interest at times to know the deflection of the parts of an airplane. The flexural deflection of the wing spars, being proportional to $\frac{Wl^3}{EI}$, will vary as l^2, if the spars are made in the same way and of the same material. Deflections of the wing truss due to the direct elongation and compression of the members also follow the same law, since the unit stress under a given load factor has been shown to be proportional to l and the total change of length of each member must therefore be in the ratio of l^2. If, however, the highest load factors actually imposed are in the ratio of $\frac{1}{l}$ the structures will deflect in a geometrically similar manner.

Deflection is perhaps most serious in its effect on the performance of the propeller, the angle of twist of the propeller blade being proportional to l in geometrically similar airplanes. A type of propeller suitable for a small airplane might, therefore, be quite unsatisfactory on a large one of the

same design, even though its calculated strength were sufficient, and tests on geometrically similar airplanes should be carried out with propellers so designed as to have a minimum of torsion.

ILLUSTRATIVE EXAMPLE.

To show how all of this work can be applied in practice, an airplane similar in general characteristics to the "Barling bomber," having a total weight of 42,000 pounds, a span of 130 feet, an area of 4200 square feet, and powered with six 400 horsepower engines, may be used as an example. Models of one-third, one-fourth, and one-fifth full size have been calculated, and their characteristics are tabulated below:

Span. Ft.	*Area.* Sq. Ft.	*Weight.* Lbs.	*Total Horse-power.*	*R.P.M.*
43	467	1560	52	2950
32	262	660	19	3400
26	168	340	8.5	3800

Obviously, the third case is impossible to realize, as the pilot's weight would be more than half of the total carried in flight, and the six engines of 1½ horsepower each would make up most of the remainder. The second case might be barely possible with 3 horsepower engines specially built for the purpose. The weight available for structure would be about 300 pounds, the area being 260 square feet, and the wing loading 2.5 pounds per square foot. The first case would be easy to realize.

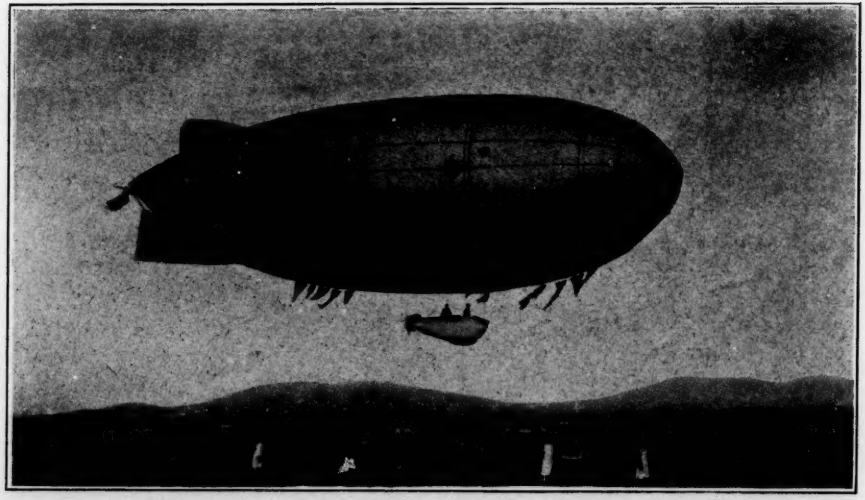
With everything considered, and the advantage and drawbacks of the light 'plane as a flying model balanced against each other, it still seems quite possible that the construction of such flying models would be well worth while in some cases, particularly if the development of large airplanes of eccentric form and arrangement is to continue, and the practice initiated by the French constructor, already referred to, may on occasion prove a profitable one elsewhere.—"Flight," September 4, 1924.

THE ITALIAN SEMI-RIGID AIRSHIP *MR.*

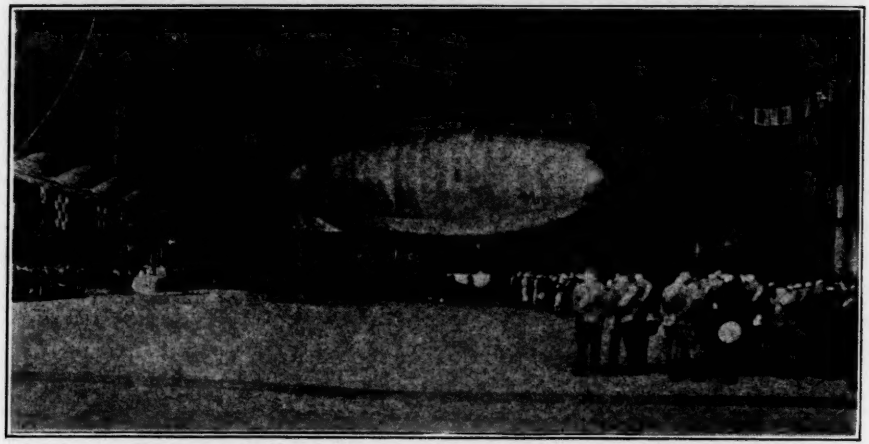
THE WORLD'S SMALLEST DIRIGIBLE.

Early in June last an interesting ceremony took place in the large airship hangar on the Ciampino aerodrome, Italy, when two semi-rigid airships, the *N.I.* (which was fully described in "Flight" for March 20 last) and the *Mr.* designed and built at the Italian Government Airship Construction Establishment, were baptized. His Excellency the Cardinal Prince Granito di Belmonte performed the religious rites before a large and distinguished gathering, Mimi Finzi acting as "Godmother" to the two airships.

One of these airships, the *Mr.*, is a new and extremely remarkable type, and we are able to give below some particulars and illustrations of this ship. It is claimed for the *Mr.* that it is the smallest airship in the world, but it is mainly remarkable on account of the following facts. Up till now it has generally been asserted that the minimum cubic capacity was possible only in the non-rigid type of airship (and except for a few experimental types, 1000 cubic meters were considered to be the practical limit), and furthermore it was stated that the semi-rigid type could not successfully be produced below 2600 cubic meters capacity. As a matter of fact, the Italian Government Airship Construction Establishment disproved this view by the production of the "S.C.A." type of semi-rigid airship (in 1921), designed by Engineer Nobile, which had a capacity of 1520 cubic meters, while at the present time they have done better still with this new dirigible, the effective cubic capacity of which does not reach 1000 cubic meters.



THE LATEST ITALIAN SEMI-RIGID AIRSHIP: THE "MR." IS CLAIMED TO BE THE SMALLEST DIRIGIBLE IN EXISTENCE. IT IS ONLY 105 FEET IN LENGTH, AND ITS CAPACITY IS 33,900 CUBIC FEET. WITH A 40 HORSE-POWER ENGINE ITS SPEED IS ABOUT 40 MILES PER HOUR.



BAPTISM OF ITALIAN AIRSHIP: THE CEREMONY, SHOWN ABOVE, OF BAPTIZING THE ITALIAN SEMI-RIGID AIRSHIP "MR." WAS PERFORMED AT CIAMPINO AERODROME LAST JUNE BY HIS EXCELLENCY THE CARDINAL PRINCE GRANITO DI BELMONTE.

When it is considered that the percentage of the useful load of an airship diminishes considerably as the capacity is reduced—a limit, of course, being reached when the airship would not be able to lift more than its own weight—the production of so small an airship as the *Mr.* is undoubtedly a technical triumph for its designer and constructors. With a capacity of 960 cubic meters the *Mr.* has a useful mean load of 450 kilograms, or 42.5 per cent of the total lift, whilst one of the smallest known non-rigid airships (the French Zodiac "Sporting" dirigible) with a capacity of 1000 cubic meters has a useful load of but 260 kilograms, or less than 24 per cent of the lift. Furthermore, while the maximum speed of this French non-rigid does not exceed 60 kilometers per hour, the *Mr.*'s speed is 65 kilometers per hour—probably a little more.

The successful results obtained with this small airship are particularly noteworthy, in that its new construction, although of an experimental nature, has proved so satisfactory that several of the structural innovations embodied in its design are to be applied to larger dirigibles, resulting in all probability in some considerable progress in airship design.

The *Mr.* was designed by Engineer Nobile, and constructed under his direction in the S.C.A. Factory, with the collaboration of other expert officials: Majors Pesco, Bruno and Biffi, and Captains Vallini and Zezi. Its production was, in a way, an experiment, one of the principal objects of its design being to operate from a base either on land or on board ship from which it can be sent out on scout work within a limited range of action. The exceptional smallness of this dirigible, besides increasing its maneuverability, in flight, assists considerably in its transportation from base to base—in fact, it should be possible to employ an automobile base, in which case the advantages and possibilities obtaining would be considerable.

It may be of interest to note here that the Spanish Navy has already effectively employed two of the S.C.A. type airships during the recent operations in Morocco, employing as base one or other of their ships with every success. In the case of the *Mr.* type, inasmuch as its capacity is less than two-thirds that of the types referred to above, such operations should present not the slightest difficulty.

From the structural point of view the new *Mr.* presents several important innovations—some of which have not as yet been disclosed. Most of our readers will remember that the fundamental characteristic of the Italian semi-rigid airships consists of the tubular metal keel frame, within the bottom of the envelope, which, while more or less rigid, is given a certain amount of flexibility by means of a series of articulated joints, and is thus capable of adjusting itself to any excessive stresses. In the *Mr.* this keel, of triangular form as in the large *N.1* type, is, we understand, absolutely rigid, the flexible joints being absent.

As in previous types, the keel extends from the nose to the stern of the envelope, and from it is suspended a neat boat-shaped car or "gondola." In the stern of the latter is mounted a 40 horsepower air-cooled engine, and it is claimed that it is so placed that a single person can manage the airship. The space within the triangular keel is utilized for the storing of fuel, water and other accessories. The nose of the envelope is reinforced by the naval "umbrella" type nose cap.

The principal characteristics of the *Mr.* are: Overall length, 32 meters (105 feet); diameter, 7.780 meters (25 feet 6 inches); capacity, 960 cubic meters (33,900 cubic feet). As previously stated, the useful load is 450 kilograms (992 pounds), but it is estimated that in the winter season this could be increased to 500 kilograms (1102.5 pounds). Under such conditions, with one person on board, the *Mr.* would have a range of 1500 kilometers (930 miles), or 25 hours' flight, and would attain an altitude of about 3000 meters (10,000 feet). In actual practice, however, this duration would be slightly less, as it is doubtful if the type of engine fitted would be capable of such continuous running. The speed is 65 kilometers per hour (40 miles per hour).—"Flight," September 4, 1924.

ASSOCIATION NOTES.

The regular annual business meeting of the Society was held on Tuesday, October 7, 1924, at the Navy Department, Washington, D. C. The following nominations were made for the calendar year 1925:

For *President:*

Rear Admiral Charles F. Hughes, U. S. N.

For *Secretary-Treasurer:*

Commander Bryson Bruce, U. S. N.

For *Member of Council:*

Rear Admiral B. C. Bryan, U. S. N., Retired.
Captain W. T. Cluverius, U. S. N.
Captain J. O. Richardson, U. S. N.
Commander S. M. Robinson, U. S. N.
Commander H. S. Howard (CC) U. S. N.
Captain Q. B. Newman, U. S. C. G.
Mr. J. F. Metten.
Mr. H. M. Southgate.

Captain Chauncey Shackford, U. S. N., resigned as a Member of the Council, his resignation was accepted with regret and Commander S. M. Robinson, U. S. N., was elected to fill the unexpired term. Captain Shackford recently took command of the U. S. S. *Marblehead.*

It was decided to hold the annual dinner of the Society in 1925. The date and particulars of this dinner will be announced to all members later.

DUES.

It is earnestly requested that any member who has not paid his dues for the current year do so at an early date, in order that the accounts of the Secretary-Treasurer may be closed.

The following have joined the Society since the publication of the last previous number of the JOURNAL:

NAVAL MEMBERS.

Alford, Clifford M., Ensign, U. S. N.
Anderson, Robert, Lieutenant, U. S. N.
Awalt, T. Young, Ensign, U. S. C. G.
Bamberger, Adolph H., Lieutenant, U. S. N.
Bard, Nathan W., Lieutenant, U. S. N.
Farmer, Claude, Lieutenant, U. S. N.
Gallagher, Edward F., Ensign, U. S. N.
Goodall, Henry W., Ensign, U. S. N.
Graham, James J., Lieutenant, U. S. N.
Herrington, Lewis B., Jr., Ensign, U. S. N.
Hopping, Halsted L., Ensign, U. S. N.
Hughes, Charles F., Rear Admiral, U. S. N.
Lee, Paul F., Lieutenant, U. S. N.
McClary, Donald B., Lieutenant, U. S. N.
Pennington, John A., Lieutenant, U. S. N.
Vylacil, Nicholas, Lieutenant, U. S. N.
Wilson, Stanley L., Lieutenant, U. S. N.

CIVIL MEMBERS.

Huey, Clarence, Material Laboratory, Building No. 22, Navy Yard, New York, N. Y.

Wadhams, A. J., Director of Research, International Nickel Corp., 67 Exchange Place, New York City.

ASSOCIATION MEMBERS.

Goeckler, Frank E., Asst. Supt. Research, Midvale Steel Co., 6508 Wister St., Germantown, Philadelphia, Pa.

Lamb, Carl J., Lieut. U. S. N. R. F., Westinghouse Electric and Mfg. Co., Lester, Pa.

Schmidt, John D., Turbine Engineer, Large Turbine Dept., Westinghouse Electric and Mfg. Co., Lester, Pa.